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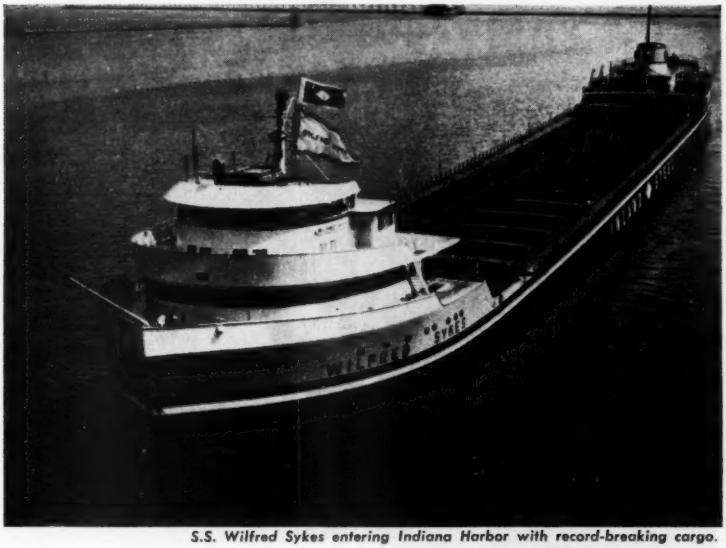

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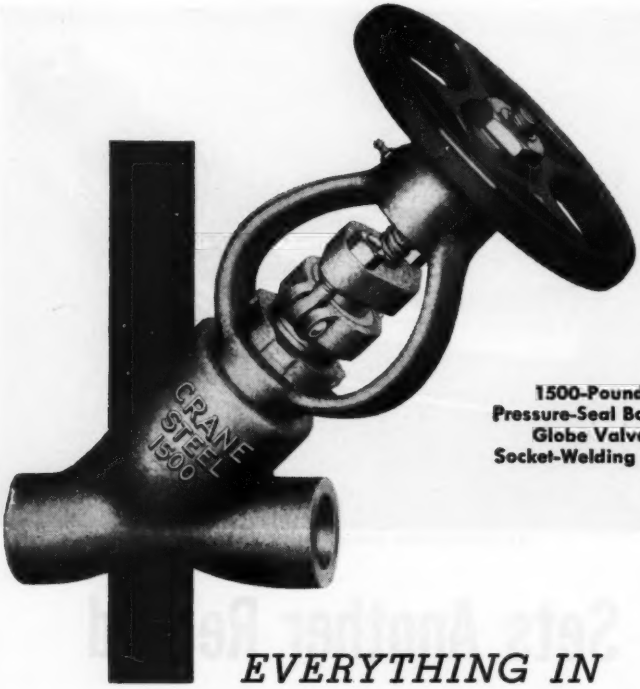

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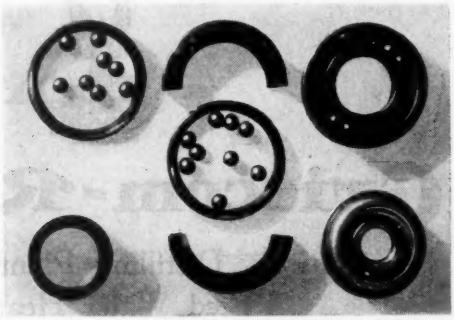

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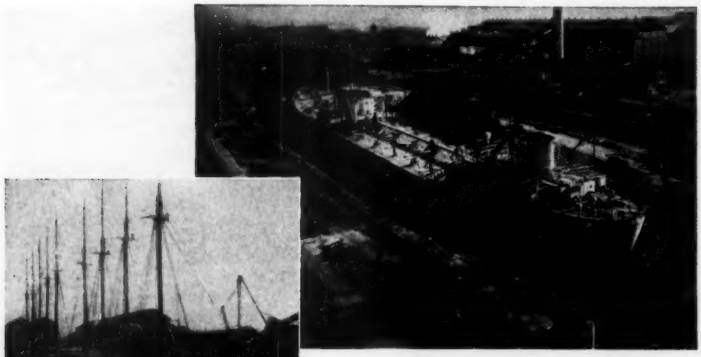

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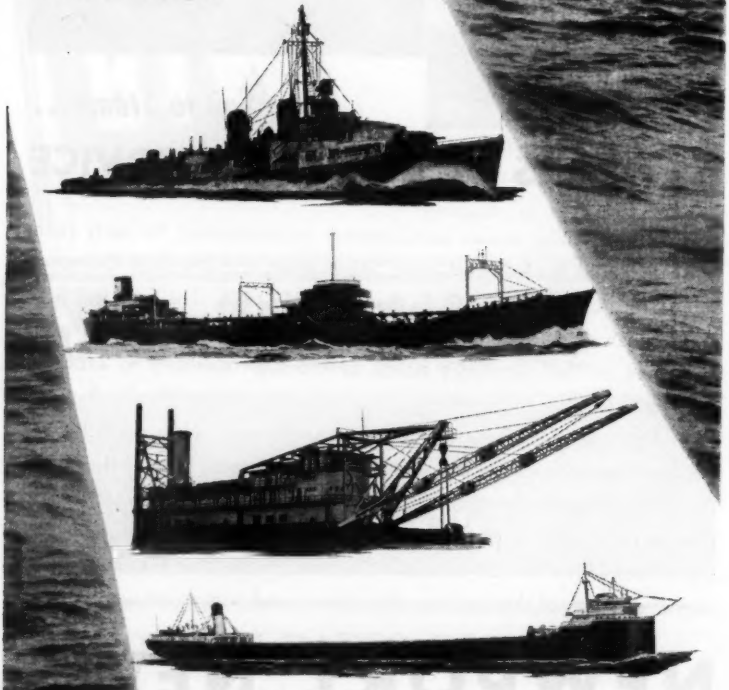

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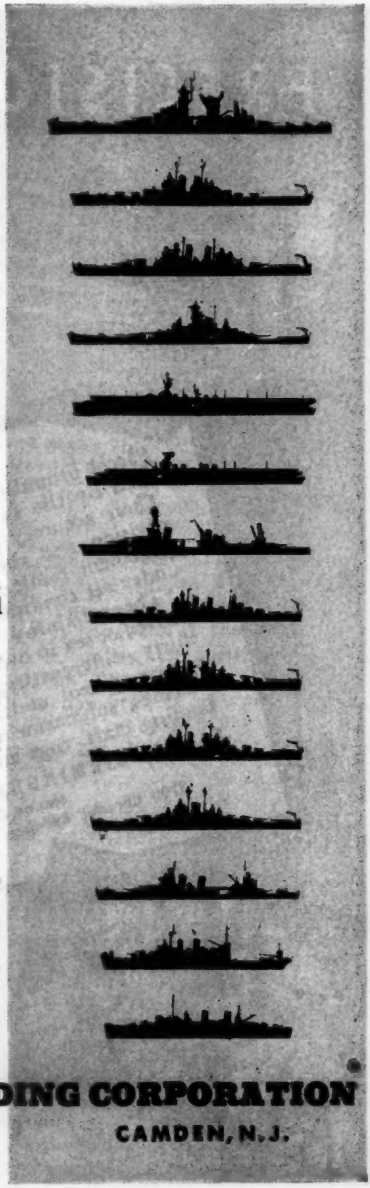

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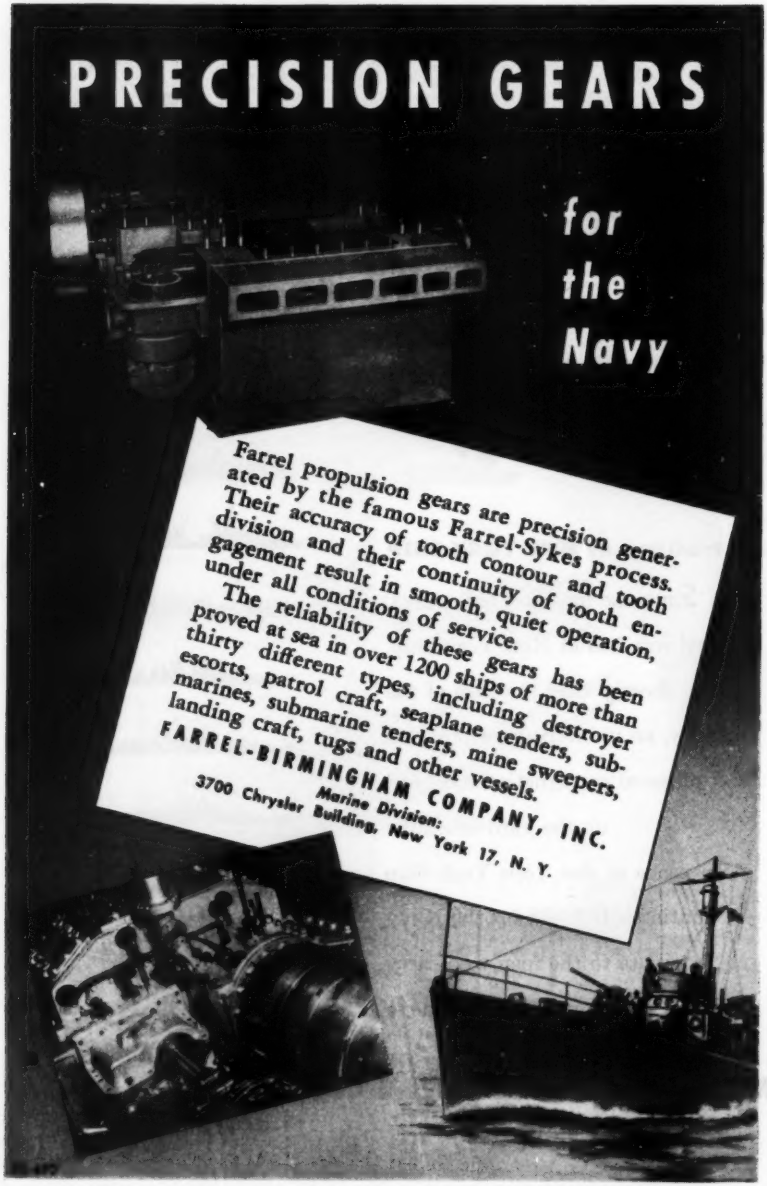
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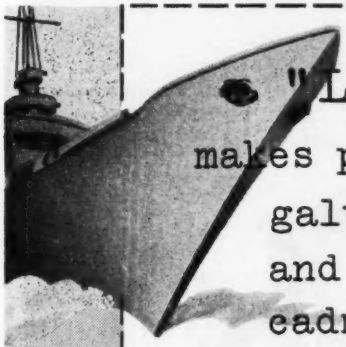

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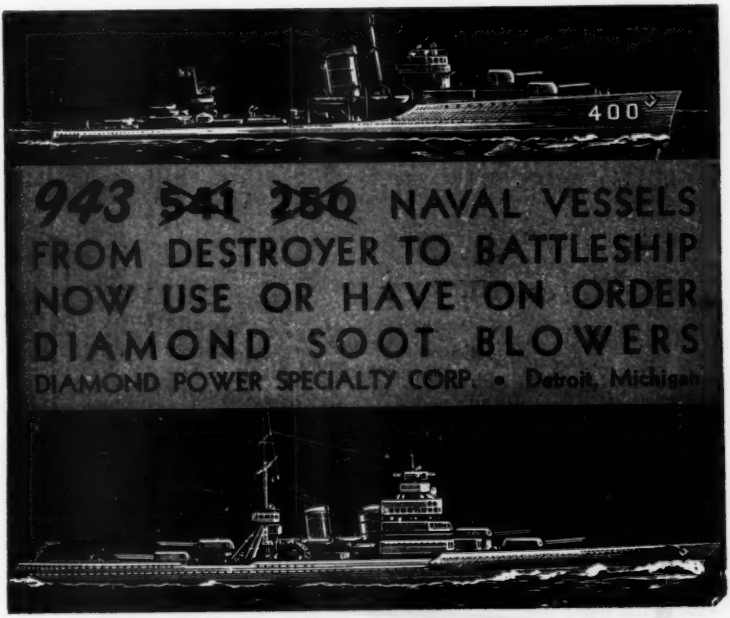
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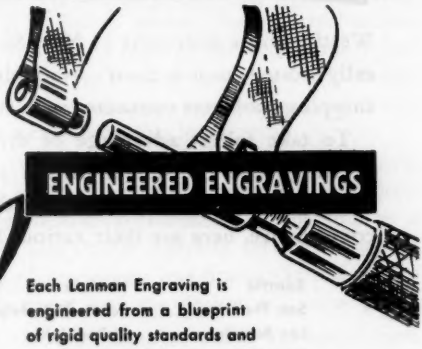

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SECRETARY'S NOTES

DD 828

When the U.S.S. *Timmerman,* DD 828 (see page 528) was launched at the Bath Iron Works Corporation at noon on May 19, 1951 a dream of naval engineers seemed closer of realization. Here is a ship of destroyer size, a high power combatant vessel in which the design engineers were given a very free hand. Better individual performance of each machinery unit, main and auxiliary, with radical reduction in weight was the double goal.

With that kind of freedom it can be expected that naval engineering will advance more on a broad front than has been possible in the past where reliable performance of the ship as a whole forced conservatism and almost complete service evaluation and acceptance of a piece of machinery was practically mandatory before it could be incorporated into the design at a new plant.

Whether the *Timmerman* becomes an active combatant unit of the fleet or a floating laboratory she should be the bridge over which naval machinery and its application advance with a Gargantuan stride.

KINGSBURY

The following letter from one of the Society's members of longest standing adds interesting additional information to the article which was reprinted on page 433 of the May issue of the JOURNAL.

ERNEST H. B. ANDERSON
266 Kimball Avenue
Westfield, N. J.

Monday, June 18, 1951

The American Society of Naval Engineers, Inc.
605 "F" Street, N.W.
Washington 4, D. C.

Gentlemen:

The article entitled "Development of the Kingsbury Thrust Bearing", reprinted in the May issue of the JOURNAL, is of special interest to me. It seems, however, that Mr. Kingsbury does not do justice, either to himself or to American marine engineers, in connection with the pioneer installation of his bearings for propeller shafts in the U.S. Navy. I refer specifically to Destroyer No. 60, U.S.S. *Wadsworth*. The contract for building this vessel was signed on October 15, 1913, and the official trials took place in May, 1915. A complete description of the vessel and the propelling machinery was printed in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. 27, 1915, at pages 640 - 662.

I also refer to the U.S. Naval Academy book entitled "Steam Turbines", a Treatise covering U.S. Naval practice, (1916) by G. J. Meyers, Lieut. Commander, U.S.N., and draw to your attention *Wadsworth* and the Kingsbury Thrust Bearings, see page 74 and Figure 63.

Mr. Kingsbury, in the second last paragraph of his paper, specifically mentions Admiral Griffin and Admiral Dyson. Both officers were Presidents of the A.S.N.E. between 1913 and 1916, and I feel assured that both officers approved the use of these thrust bearings in this destroyer.

In conclusion, I can only add that the machinery installation of *Wadsworth* was an engineering achievement, designed and built under the supervision of an American marine engineer and naval architect; and without doubt, it was this gentleman whom Mr. Kingsbury *cannot now identify*.

Finally, the references to "Mitchell" are of little or no concern, and the correct spelling of the name in question is "Michell" (see A.S.M.E. Paper).

I have purposely refrained from bringing in the names of engineers who took leading parts, for there are many men who know about this matter of the propelling machinery of this remarkable Naval vessel.

Yours faithfully,

ERNEST H. B. ANDERSON
Associate and Civil Member,
1907 to date

Annual Banquet

The Secretary proposes to discuss the following questions at the next annual meeting in October in order to get some member reaction. Decisions, of course, will be made at the appropriate time by the Council and by the Banquet Committee.

1. Shall attendance be limited to a smaller number than is now permitted?

The Statler Hotel has the only banquet arrangement in Washington which can accommodate a really large group. As those who attend the banquet know, the two large connecting rooms have been used but as the attendance increased more and more tables have had to be placed outside in the foyer and corridors. This year at least 300 persons were seated outside. In spite of that there were only scattered complaints. If the rule which was followed this year is continued, 1572 acceptances are possible. If seats within the main rooms *and* with a view of the speaker's table determine capacity, the cut-off point will be about 1200.

2. If attendance is limited shall it be done by cutting off when the limiting number of applications is received or by restricting attendance to members only?

For some years each member has been allowed one non-member guest. Attendance has run about 60 percent members and 40 percent non-members. It is believed that elimination of non-members would hold attendance down but this is not assured. Most of those who attend are eligible for membership and it is possible that this restriction might augment the membership without giving the relief to banquet attendance which is sought.

3. Shall favors at the banquet be continued?

4. Shall the cost of the banquet continue to be less for members than for non-member guests?

The above questions come up each year and are settled by the Council or by its appointed committee. They are so surely recurrent that any opinions of members in general would be very helpful. Anyone with sufficient interest in the subject to write a letter before the annual meeting is urged to do so.

The Journal

Again, a plea is made for comment and constructive criticism on the format and contents of the Journal. We are still carrying

the tear sheet in the back so that it is reasonably easy for anyone to suggest subjects for articles.

MEMBERSHIP

The membership list continues a steady but very slow growth. It can stand the added impetus of at least one new member for each present member. After reading "The Second Mile" which starts on the opposite page, and which reading is strongly recommended, some of the intangible benefits of membership in The American Society of Naval Engineers will be better understood.

Several years ago "The Second Mile" was reprinted in the JOURNAL. *Such a classic cannot be overcirculated. It thoroughly merits this second* JOURNAL *appearance.*

THE SECOND MILE

A REVISED VERSION

WILLIAM E. WICKENDEN

WILLIAM E. WICKENDEN

Born, Toledo, Ohio, December, 1882; graduated with honors, Denison University, 1904; graduate student and instructor, University of Wisconsin, 1905-09; member of electrical engineering staff, Massachusetts Institute of Technology, 1909-18; personnel manager, Western Electric Company 1918-21; assistant vice-president (relations with colleges), American Telephone and Telegraph Company, 1921-23; director, Investigation of Engineering Education, 1923-29; president, Case Institute of Technology, 1929-47; president, American Society for Engineering Education, 1933-34 and its Lamme Medalist, 1935; president, American Institute of Electrical Engineers, 1946-47; honorary degrees from nine institutions; died September, 1947.

This revised version of "The Second Mile" was published by the Engineers' Council for Professional Development.

"Whosoever shall compel thee to go one mile—go with him twain." I am not sure that I should dare to choose this as a text for a talk to engineers, such is the present state of our biblical illiteracy. However you will recognize it from the Sermon on the Mount as a counsel of perfection, good advice in the form of a paradox which emphasizes a profound truth by an apparent denial of common sense.

A preacher, who was reproached for straying rather widely from his text, replied "A text is like a gate, it has two uses; you can either swing on it, or you can open it and pass through." Let us pass on through.

Every calling has its mile of compulsion, its daily round of tasks and duties, its standard of honest craftsmanship, its code of man-to-man relations, which one must cover if he is to survive. Beyond that lies the mile of voluntary effort, where men strive for excellence, give unrequited service to the common good, and seek to invest their work with a wide and enduring significance. It is only in this second mile that a calling may attain to the dignity and the distinction of a profession.

IS NOT EVERY OCCUPATION A PROFESSION?

There seems to be an illusion abroad that any calling may claim recognition as a profession by merely willing it so and by serving notice to that effect on the rest of the world. It is supposed to help a lot, too, if you can invent for your occupation some mysterious-sounding name derived from the Greek. One reads, for example, of a group of barbers who elect to be known hereafter as "chirotonsors," in order to raise the prestige of their "profession." Why not, when you have cosmetologists right in the same shop? The truth seems to be that as soon as any word acquires a halo of distinction, every one wants to claim it, and the unique value of the word is quickly destroyed by indiscriminate usage. When one scientist observed what the advertising fraternity has done to the word "research," he remarked dryly that we now use that word to mean so many things we shall soon have to invent another word to mean research. The ambition to dignify honorable work is laudable, but there is much seizing after the form and letting the substance escape, which would be ludicrous if it were not pathetic.

A prominent English churchman once remarked facetiously that there were three sorts of Anglicans—the low and lazy, the broad and hazy, and the high and crazy. It seems to be much the same among engineers in our thinking about our profession. We have a low church party which holds that status and titles are of little consequence; so long as the public allows us to claim them not much else matters if the engineer does an honest day's work. The broad church party is all for inclusiveness; if business men and industrialists wish to call themselves engineers, let us take them in and do them good, not forgetting the more expensive grades of membership. The high church party is all out for exclusive definitions and a strictly regulated legal status; in their eyes, what makes a man a "professional" engineer is not his learning, his skill, his ideals, his public leadership—it is his license certificate.

WHAT PROFESSIONS HAVE IN COMMON

Of professions there are many kinds; open professions like music, to which any man may aspire within the bounds of his talents, and closed professions like medicine which may be entered only through a legally prescribed process; individual professions like painting and group professions like law, whose members constitute "the bar," a special class in society; private professions like authorship and public professions like journalism, artistic professions like sculpture and technical professions like surgery; ameliorative professions like the ministry and social work, and professions which safeguard social institutions through a technique of destruction, like the army and navy. Despite all these differences of pattern, certain characteristic threads run like a common warp beneath the varying woof of every type of professional life and endeavor.

If one searches the authorities for definitions of a profession he will probably find four kinds. One is likely to hold that the determining quality is *an attitude of mind*, that an altruistic motive can lift any honorable calling to the professional level. A second may say that it is a certain *kind of work*,

one requiring special skill on a high intellectual plane. A third may state that it is a special *order in society,* as the bar, the bench or the clergy. Still others insist that no work can be professional without a *confidential relationship* between a client and his agent, as that of patient to physician, litigant to lawyer, etc. None of these definitions is self-sufficient. Taken together, like the legs of a table, they give a profession a stable base of support.

THE INDIVIDUAL AND THE GROUP

Suppose a man were to say, "I'll be a professional, but I'll do it by myself; I hate organization dues and duties." One would be tempted to reply, "So you'll take the head, without the body which nourishes it." Or suppose another were to say, "Oh! I'll join all right, but I'll never do anything about it," to which a fair answer would be, "So you'll grasp the form and let the substance go." Full professional life can not be achieved in isolation; like religion and the church, it needs a culture medium in which to grow and flourish. Some aspects of professional life are essentially individual, while some are essentially group relations.

What is the distinctive mark of the professional man? First, we may say that it is a *type of activity* which carries high individual responsibility and which applies special skill to problems on a distinctly intellectual plane. Second, we may say that it is a *motive of service,* associated with limited rewards as distinct from profit. Third, is the *motive of self-expression,* which implies joy and pride in one's work and a self-imposed standard of excellence. And fourth, is a conscious *recognition of social duty* to be fulfilled among other means by guarding the ideals and standards of one's profession, by advancing it in public understanding and esteem, by sharing advances in technical knowledge, and by rendering gratuitous public service, in addition to that for ordinary compensation, as a return to society for special advantages of education and status.

Next, what attributes mark off the corporate life of a group of persons as professional in character? We may place first *a body of knowledge* (science) and of *art* (skill) held as a common possession and to be extended by united effort. Next is *an educational* process based on this body of knowledge and art, in ordering which the professional group has a recognized responsibility. Third is *a standard of personal qualifications* for admission to the professional group, based on character, training and proved competence. Next follows a *standard of conduct* based on courtesy, honor, and ethics, which guides the practitioner in his relations with clients, colleagues and the public. Fifth, we may place a more or less formal *recognition of status,* either by one's colleagues or by the state, as a basis for good standing. And finally, there is usually *an organization* of the professional group, devoted to its common advancement and its social duty, rather than to the maintenance of an economic monopoly.

HOW PROFESSIONS CAME INTO BEING

The traditional professions of law, medicine and divinity had a common fountainhead in the priestcraft of antiquity. What is professional in modern technical callings such as engineering can be traced back only as far as the mediaeval craft and merchant guilds, which arose out of the breakdown of feudalism. Before strong central and local governments were developed to

lend the protection of army or navy to far-flung commerce, voluntary groups of merchants had to band together for their own protection, as in the cities of the Hanseatic League along the shores of the Baltic Sea. The various crafts likewise found it necessary to join together in guilds for mutual protection and regulation. If you have an opportunity to see a performance of Wagner's opera *Die Meistersinger* you will not only hear much magnificent music but also witness an unforgettable picture of guild life in its mediaeval home.

There being no other effective authority, the guilds took over the regulation of the hours of labor, the observance of holidays, the length and content of apprenticeship, the wage system, the standards of workmanship and the quality of goods. The guilds also tested the progress of novices, apprentices and journeymen, and finally admitted them to the ranks of the masters with imposing ceremonies, of which college commencements and inaugurals are the most picturesque survivors in our modern day.

The guilds naturally took unto themselves considerable monopolies and privileges. As the cities gradually grew strong they usually recognized the guilds and gave them a considerable share of civic responsibility. To this day The City and Guilds of London are a single corporate unit. The church, too, lent its blessing in keeping with a philosophy which looked upon society as a commonwealth divided into divinely ordained functions and not as a mere aggregate of individuals. In short, guild life became a highly developed form of citizenship which centered around occupations rather than politics.

Many features of this distinctive type of citizenship are perpetuated in our modern professional bodies. The public grants to a profession more or less tangible monopolies and self-governing privileges, in consideration of which the profession engages to admit to its circle only men of proved competence, to guarantee their trustworthiness, to insist on the observance of ethical relations and practices, and to protect the public against bungling and extortion.

WHAT A PROFESSION MUST GUARANTEE

The self-policing responsibilities of a profession in our modern world arise in large measure from the fact that so many of the occasions which call for professional service are human emergencies in which the usual axioms of business, such as "competition is the life of trade" and its legal doctrine of *caveat emptor,* "let the buyer beware," simply break down. When a baby is about to be born or an appendix must be removed, you do not want to drive a smart bargain at your own risk. What you want are credentials that will assure you that the job is in competent hands.

When a layman comes face to face with the complex and often terrifying specialization of professional skill and knowledge, he is likely to be baffled or easily misled. If you have a problem of mental illness in your family, how can you be sure you are dealing with a qualified psychiatrist and not with a plausible but unscrupulous quack? To protect you in these emergencies the public wisely puts the burden of guaranteeing at least minimum standards of competence and ethics on the profession itself. The physician you can trust is the one who is recognized as well qualified and reputable by his brother physicians of good standing; the same with lawyers, dentists, architects and engineers.

The state may implement the obligation of a profession to guarantee competence by designating a group of its members to conduct professional examinations and to issue public licenses to those who pass them successfully, or it may leave professional bodies free to issue their own credentials. Both of these practices may exist side by side, as in the realm of medicine, where the state licenses general practitioners through a board of examiners, while the various groups of specialists have voluntary organizations which examine and certify physicians seeking recognition in their respective fields. In the end, it comes down to the same principle—a profession must guarantee to the public the trustworthiness of its practitioners. In return, the public protects the profession from the incompetent judgment of the layman by a privileged position before the law.

IS TECHNICAL SKILL ENOUGH?

Professional status is therefore an implied contract to serve society, over and beyond all specific duty to client or employer, in consideration of the privileges and protection society extends to the profession. To possess and to practice a special skill, even of a high order, do not in themselves make an individual a professional man. Mere technical training, at any level, is vocational rather than professional in nature. The difference between technical training and professional education is no simple matter of length—any difference of two years, or four, or six; nor is it a mere matter of intellectual difficulty. It is rather a matter of spirit and scope. More specifically, it can be described as an overplus beyond the knowledge, however intricate, a man needs to master his daily tasks.

A surgeon, for example, needs to be extremely skilled in tying knots, which he learns with the other skills of his craft by technical training and prolonged practice, but the overplus in his education which marks him as a professional man is the long process of study, observation and reflection which gives him deep insight into the human organism and its hidden forces of disorganization and recuperation. Professional education for the lawyer means more than training him to draw contracts, or to draft wills and trust agreements, or to prepare briefs and try cases. It means also the study of the history and philosophy of human relations and social institutions out of which the law has grown as a deposit of the age-long experience of mankind. Likewise for an engineer, technical training aims at skill in applying mathematics and the physical sciences to concrete problems of design, construction and operation, but professional education looks beyond to philosophic insights into the relations of mathematics and science as modes of universal human experience and to competent understanding of the social and economic forces set in motion by technological achievement.

The overplus in professional education, in short, is that which enables a professional man to view his work not only as a skilled service to a client, but also in terms of its consequences for society. An engineer, for example, develops a labor-saving process and recommends its adoption; does he see in this act only an immediate saving in the cost of production, and assume that this is adequate justification in itself? Or can he perceive the sequence of effects which will be felt in the lives of individual workers, of the organization which employs them, of the community in which it functions, of the

markets which it supplies, and of the wider sector of society which it ultimately serves? In the answer to these questions there is wrapped up much of the difference between a high-grade technician and an engineer of true professional stature. Every professional body counts on its rosters many men who are little more than technicians, and it is well that they are included since professional development comes so largely through association and indoctrination. But no professional body can be strong and effective unless it contains a substantial nucleus whose intellectual attainments far exceed in depth and breadth the technical demands of its practice.

PROFESSIONAL OBLIGATIONS

The ethical obligations of a profession are often embodied in codes and enforced by police powers. Even when no written canons exist, as in the artistic professions, unwritten usages and standards exercise a powerful guiding and restraining influence. The physician and the lawyer are bound by explicit obligations. Each must keep inviolate his client's confidence, serve his client's interests exclusively, and stay within strict bounds of procedure. Woe betide the man who oversteps these obligations; if the courts do not deal with him the organized profession will. As engineers, our codes are less tangible and the means of enforcement less explicit, in proportion as our duties are less definable, but our ethical obligations are no less binding morally.

Codes and police powers alone do not suffice to sustain the personal and corporate obligations of a profession any more than statutes and courts alone can assure the healthy life of a community. Equal importance attaches to the state of mind known as professional spirit which results from associating together men of superior type, and from the adherence to a common ideal which puts service above gain, excellence above quantity, self-expression above pecuniary motives and loyalty above individual advantage. No professional man can evade the obligation to contribute to the advancement of his group. His skill he rightly holds as a personal possession and when he imparts it to another he justly expects a due reward in money or in service. His knowledge, however, is to be regarded as part of a common fund built up over the generations, an inheritance which he freely shares and to which he is obligated to add; hence the duty to publish freely the fruits of research and to share the advances in professional technique. If the individual lacks the ability to make such contributions personally, the least he can do to pay his debt is to join with others in creating common agencies to increase, disseminate and preserve professional knowledge and to contribute regularly to their support. That is the purpose to which a large share of the membership dues of our professional societies is devoted.

Too many engineers exhibit an unenlightened and petty attitude on these matters; mature men who complain that the direct returns to them of the researches and publications of a professional society are not worth the annual fee, and young men who grumble because membership does not lead to direct preferment in rank and salary. Shame on us! Do we look with envy on the high prestige of medicine or surgery? Then let us not forget that this prestige has been won not merely through personal skill and service, but not less through magnificent contribu-

tions to human knowledge without profit to the seekers and with incalculable benefits to all mankind. Do we covet leadership on a par with the legal profession? Then we do well to remind ourselves that a profession exceeds a technical vocation in that measure of personal development and powers of leadership which fit men for places of influence in the community.

MARKS OF RETARDED PROFESSIONAL DEVELOPMENT

Measured by standards such as these, many men who call themselves engineers and who have proved themselves competent in accepted technical practices, have not attained a real professional stature. Some are victims of a deficient education, not in the sense that school and college failed to teach them all they would ever need to know, but rather failed to inculcate a taste and a capacity for continued learning under self-direction. They are usually the men who have let their scientific training slip away after they have mastered a specific job, who have been unable to surmount the routine of early experience and have gradually grown content with mediocrity. Some of the difficulty may be inherent in the operating routine so often associated with an engineer's work. There is much in the daily work of a physician, a lawyer, or a minister of religion which compels him to be a life-long student. In peacetime, the young army officer is likely to spend an average of one year in six going to school. The student habit is less often a mark of the engineer, and in some degree this is but natural to the man of action as distinct from the man of reflection, but far too many are content to leave all growth after their college days to the assimilation of ordinary experience, without sustained intellectual discipline of any kind. If these deficiencies exist, they are not solely a reflection on the individuals involved, but also on the professional body. One of its obligations to its younger members is to give effective stimulus and guidance to their growth.

WHY NOT REQUIRE LONGER TRAINING?

There is a school of thought which has two quick and ready remedies for all the ills and shortcomings of the engineering profession. One is to keep the boys longer in college and to compel them to cover courses in both liberal arts and engineering. The second is to limit strictly the use of the title "engineer" to men who have obtained a public license. One need not quarrel with either the aims or the means; so far as they go both are good, but they cover only the first mile.

Registration will probably always be a qualifying standard rather than a par standard for the engineering profession. By its nature, it cannot be a standard of distinction. It will go far toward keeping the wrong men out of the profession, but it will serve only indirectly to get the right men in. Beyond it lies a second mile of growth and advancement for which effective guidance, incentives and rewards can be provided only within the profession itself.

The proposal to compel all engineering students to remain six years or more in college in order to complete combined courses in liberal arts and in engineering is attractive in theory but unworkable in practice. Some young men should do so, but the majority will not. Those who do are likely to find that the advantage gained comes quite as much, or even more, from sharing the life and spirit of two divisions of

education with differing ideals and traditions, as from a more extended range of studies. The advantage of breadth, so gained, must be weighed against the depth which might result from devoting the same length of time to a combination of undergraduate and postgraduate training. No combined program in arts and engineering requiring more than four years can be made compulsory in the face of the demands of industry, the attractive terms offered to four-year graduates, and the wide range of engineering responsibility, nor is it likely to succeed so long as the typical student engineer, as graduation approaches, shows so unmistakably that he is fed up for a time with formal teaching and study and craves action and experience.

Much has happened in late years to strengthen the belief that possibilities in the mile of voluntary advancement are more hopeful than any lengthened mile of compulsory discipline. Growth in postgraduate enrollments in engineering colleges in the prewar years was especially impressive, with the ratio of master's to bachelor's degree rising above one to ten and that of doctorates nearing one to a hundred. What was particularly noteworthy was that so many of these advanced students were pursuing interests and needs which they had discovered for themselves in their early professional experience, rather than a further discipline—however ideal—which others had imposed upon them.

DOES AN ENGINEER NEED HIS PROFESSION?

The engineer needs his profession for his personal advancement. That is the purpose which brought it into being.* He needs it most at the beginning of his career. Perhaps you have heard the wisecrack on the bringing up of the American boy, "When my father and my mother forsake me, then the Boy Scouts will take me up." Just substitute *alma mater* and *professional society* in the right places. Young men need for their advancement wider sources of information, more varied personal associations, broader stimulation to achievement and less formal contacts with their seniors than they usually find in their daily jobs. They also need earlier outlets for their organizing and executive abilities—something on a pilot plant scale like the campus activities of college life. They can gain much from outside recognition. As men mature they come to value professional rewards—friendships, recognition, responsibility, pride in belonging, evidences of distinction, etc.—no less and often more than money rewards. These are the durable satisfactions of life.

The engineer, in a society based largely on group relations, needs his profession to safeguard his occupational and economic welfare. He needs protection against unethical competition, against indiscriminate use of the title "engineer," and against all influences which might undermine public confidence in his integrity and competence. He needs protection against those who assume that he is "just another employee" and against sub-professional groups seeking to act for engineers in the process of collective bargaining. He needs protection against the levelling influences of unionism and of civil service. He needs the benefits of prestige built up through group publicity. He

* The first professional society, *The Institution of Civil Engineers*, was organized in London. The official account of its founding begins, "It was toward the end of the year 1817 that a few gentlemen, then beginning life, impressed by what they themselves felt were the difficulties young men had to contend with in gaining knowledge requisite for the diversified practice of engineering, resolved to form themselves into a society."

needs a collective instrument for shaping public policy in the realm of his responsibility. It is true that a professional organization is primarily a moral agency and not in itself an economic or political pressure group, but in the long run moral agencies are the more powerful and enduring.

The engineer needs his profession because of his stake in the advancement of knowledge and technical skill. Who has a larger stake, and who stands to gain more through pooling of effort? The ancients washed occasional nuggets of useful knowledge out of the gravel of common experience by individual placer mining, but science began to get on only when men organized agencies to collect, preserve and disseminate knowledge and to mine the hidden veins systematically. Otherwise, there would be no profession of engineering. Industrial research, invention and the patent system, with all their incalculable benefits, tend to canalize knowledge and know-how in restricted channels. The general interest and professional advancement alike require that the reservoir of free knowledge, stored by centuries of untrammelled research, be constantly renewed. Few engineers can do much about it alone. Collectively their capacity to advance knowledge is beyond calculation.

Most of all that has been set down above has an idealistic base. If one insists on being a Philistine, it can easily be brushed aside with a "So what?" Some consider this the natural reaction of a man who day by day must give hardheaded answers to the questions "Will it work?" and "Will it pay?", and whose work leaves no room for wishful thinking. But look for a moment under the surface. By common consent, the quality most universal and indispensable among engineers is integrity; this is essentially something moral or idealistic. Ranking almost equal with integrity is devotion to duty; given a job, an engineer will see it through, come hell or high water; and so on through the whole catalog of the engineer's distinctive virtues. Why is he so? Because a boss or a time-clock is policing his efforts? Or a money incentive drives him? Or hoped-for applause urges him on? Well, hardly! Because he has had a soldier-like training and indoctrination? In some small measure. Because he has a tradition to uphold? Yes, no doubt. Or is it because of something within himself to which he dare not be disloyal, and a faith between himself and his colleagues—unspoken perhaps—which he dare not betray?

In matters of social concern is the engineer just a materialist, promising to save civilization by flooding it with gadgets? No more than other men who bear high responsibilities in the industrial order. He is not only an engineer but a citizen, not only an individual but one of a corporate group with a major contribution to make to the common good. Naturally, he has a concern that the prevailing social attitudes and public policies shall favor rather than hinder that contribution. Amid the present Babel of economic confusion he has a message to proclaim in clear and ringing terms—that our economic ills cannot be solved by division until they are first solved by multiplication, that the creation of wealth in a democracy is the job of free enterprise, and that the public has a far greater stake in stimulating technical progress than in policing prices and profits. The dimensions of this issue are no longer national, they are world-wide. Shall the engineer raise his voice from an individualistic soapbox, or through the amplifier of professional organization?

Millions of individual, unrelated efforts will not add up to the future that invites our profession. This is no time for engineers to wrap themselves in the mantle of isolation; let us get together and be about our common business.

A LOOK AHEAD

Let us risk a look into the next 50 years, which our present student engineers are to share in shaping. The climax of man's effort to subdue nature, to shift labor from muscles to machines, to make material abundance available to all, and to extend a high civilization into the backward areas of the world may well fall within their lifetime. After that, perhaps human interest may shift from work to leisure, from production to enjoyment, from economic progress to culture and from industry to art. Who knows? In the meantime, however, it seems inevitable that industry will be extended on worldwide lines, production will grow more scientific, research will expand, and engineers will multiply accordingly.

Engineers will find their way into every field where science needs to be practically applied, cost counted, returns predicted, and work organized systematically. They will be called upon to share the control of disease with physicians, the control of finance with bankers, the bearing of risks with underwriters, the organizing of distribution with merchants and purchasing agents, the supplying of food with packers and purveyors, the raising of food with farmers, and the operation of the home with housewives. In few of these new fields, if any, will engineers be self-sufficient; to be useful they must be teamworkers; and they must be prepared to deal with "men and their ways," no less than "things and their forces."

The engineering profession, it seems equally evident, will bear much heavier responsibilities in civic and economic affairs. It cannot afford to become either a narrow caste of highly skilled technicians or a free-for-all alumni association of engineering graduates. It will probably never be able to define its boundaries precisely, nor become exclusively a legal caste, nor fix a uniform code of educational qualifications. Its leaders will receive higher rewards and wider acclaim. The rank and file will probably multiply more rapidly than the elite, and rise in the economic scale to only a moderate degree.

The engineer's job will be so varied, and will change so fast, and his tools will so increase in variety and refinement with the advance of science, that no engineer can hope to get a once-and-for-all education in advance.

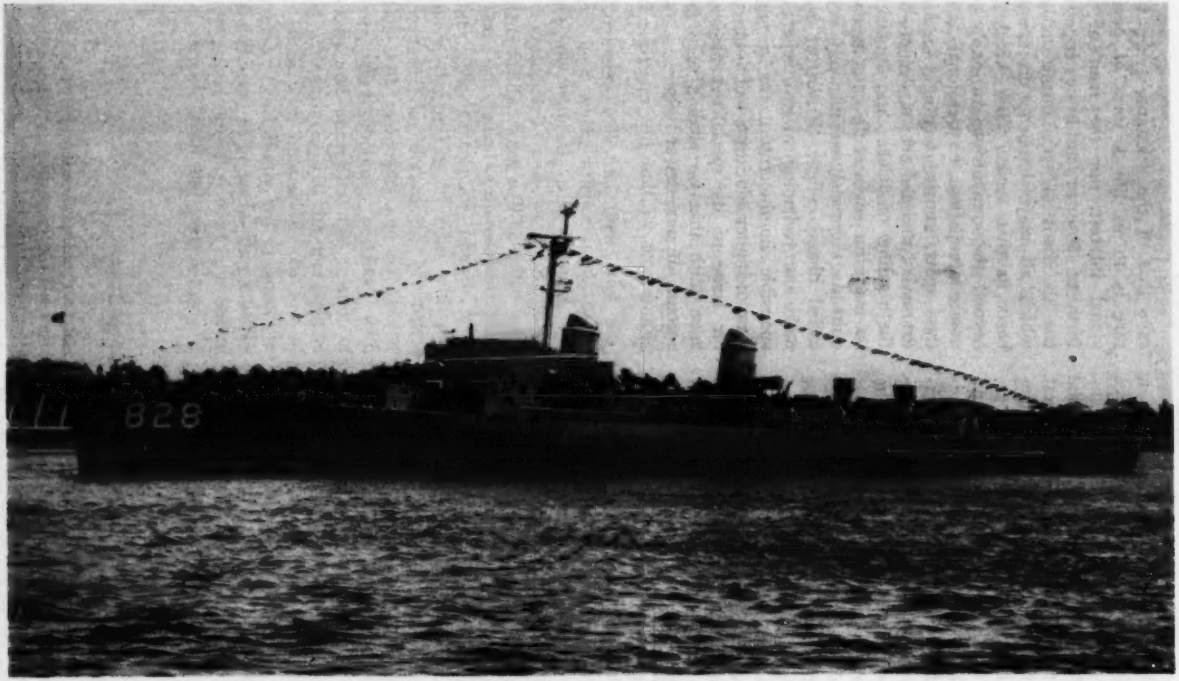
We should cease to think of education as a juvenile episode. We should expect to re-educate engineers either continuously or at intervals throughout their active careers. Once the needed means of after-college education are provided in ample degree the engineering colleges could broaden the scientific and humanistic bases of their curricula, cut down on early specialization, relieve overcrowded schedules, inspire independent work, and show the world the best balanced and best integrated of all modern disciplines.

COMPETENCE AND CULTURE

There are undoubtedly some who feel that the cultural and spiritual interests of society are menaced by a greater dominance of technological education. I am unable to share these fears. We of the engineering schools have no quarrel with liberal education. We recognize that there are great numbers of young people whose career purposes come to a focus late rather than early in adolescence and who do well to lay first their foundations of culture and social understanding before attempting to build up some special competence.

We believe, however, that there are a great many more young people than are now provided for whose career aims can be brought to a focus late in the secondary period and who will do best to lay first the foundations of competence, then to erect on them a superstructure of social understanding and personal participation in cultural activity. If we were to criticize the traditional emphasis of the liberal arts, we should do so on the ground that the preservation and advancement of culture and social ideals, except for certain highly trained specialists, are not the obligation of a special elite in today's society, but one which rests equally on men and women in all occupations and social groups.

We are not indifferent to culture, save that of the dilettante type. Culture is to us not a form of professional interest, nor the fruit of any form of pose or academic exposure, but the fruit of spontaneous activity which all may share on an amateur basis in that second mile which lies beyond the compulsions of one's economic occupation. Expressional activities—sport, music, writing, speaking, dramatics, and the arts of design—also the reading of books, are flourishing on many an engineering campus today quite as vigorously as in many a so-called liberal college. If destiny is to make our technological institutions responsible in the future for a major stem of higher education, and not merely for some of its specialized phases, I have faith that we shall give a good account of our stewardship.



U. S. S. *Timmerman* (DD828)
Built by Bath Iron Works Corporation
Launched May 19, 1951

PART ONE

ORIGINAL ARTICLES

The articles which the Editor, acting under the supervision of the Council, has accepted for publication in Part One are the original works of the authors.

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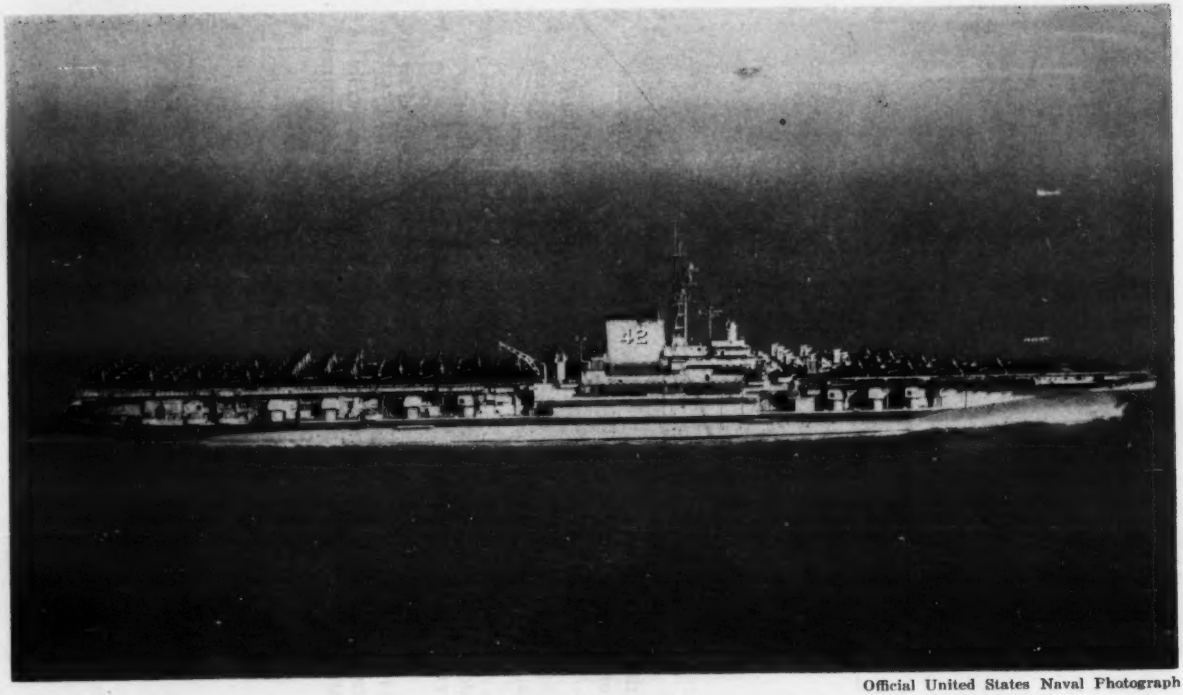


Official United States Naval Photograph

A NEW ESSENTIAL FOR VICTORY AT SEA

LIEUTENANT G. P. STEELE II., U. S. NAVY

THE AUTHOR

graduated from U. S. Naval Academy in 1944 with the Class of 1945. Served in submarines since that time. Made two war patrols during World War II in the Southwest Pacific Area in USS *Becuna* (SS319). Attended Electronics Materiel School, Treasure Island, Calif. (six months' course). Served as electronics officer of a submarine for about three years total as well as in all of the other departments. Was Electronics Officer, Staff, Commander Submarine Squadron Eight from 1950 to 1951. At present an instructor at the Submarine School, New London, Conn., in the Tactical Department.

Is the Navy being as slow to fully adopt electronics as it was steam? In 1807 Robert Fulton proved that the steamboat was practicable. In 1907 Lee DeForrest patented the first vacuum tube. Both events were of revolutionary significance to the United States Navy. Both events were slow to make their effect felt in the service. Still, have we been more receptive to new ideas in our century than our great-grandfathers were in theirs?

In 1850 you could walk aboard any ship of the Navy and get it underway without a flicker of the steam pressure gauge. Commanding officers had been brought up in the age of sail which stretched back to the earliest history of mankind. Many barely tolerated the dirty, cranky steam engines that soiled the holy-stoned topside with a layer of soot and cinders. Few took the trouble to learn the intricacies of the new infernal machinery. The old sea dogs didn't want to learn any new tricks—and so they had started the Engineer Corps in 1836. About the middle of the 19th Century "many distinguished officers wagged their heads wisely at the sight of steam ships of war and predicted that they would never withstand the shock and fire of battle." [1] In 1850 steam was on the scene but it had by no means been fully accepted and exploited.

In 1950 our commanding officers had all been brought up in the pre-electronic age. But just as the Mexican War showed that steam had a place, so

[1] The Monitor and the Navy Under Steam—E. M. Bennett, Houghton, Mifflin & Co., 1900.

World War II had demonstrated that electronics had a place. It was a rare commanding officer that had any but the most cursory knowledge of electronics. As they were a hundred years before, officers were given special training in the new skill and assigned duty in it. Without the wisdom of yet another hundred years we may argue that our record as of 1950 was not nearly so damning as that of 1850. Today most commanding officers have learned to lean heavily on electronics, to demand more and better equipment, and to expect high performance from their electronic equipment and personnel. The Navy is pouring millions of dollars into electronic research and development as well as into the procurement of new "hardware." Interest in electronics has burgeoned as never before. But there are plenty of dark spots on our record, too.

It is a short record. Electronics in quantity has existed for the regular naval officer for the short span of only ten years. In ten years we have traveled far and fast. We have come from the day of the human eye to the day of the electronic eye; from the lookout to the radar; from the sextant to the Loran; from the visual identification signal to the electronic IFF. Our progress has been so fast that problems have not always been solved as they came up. Some have simply climbed on board and come along, growing bigger daily. The time has come to face and solve them.

Let us not fool ourselves. The performance of the Navy's electronic equipment in 1950 was not satisfactory. One fleet commander last year estimated that ten percent of his electronic equipment was always out of commission and that another fifty percent was unreliable. Altogether, sixty percent of the electronic equipment in service in that fleet was in poor condition! And despite the untiring efforts of all concerned that black picture had not been appreciably improved for the previous two years. This problem of electronic maintenance is vital and it is urgent.

As is the case with most problems, people are at the root of this one. Electronic repair personnel procurement, training, distribution, and direction is a large part of it. Equipment design and procurement is another large part. The design and procurement of the electronic components, the resistors and the tubes, and the other myriad of complex items, is yet another. But above all, the education of the service as a whole in electronics is essential to a happy resolution of our difficulties. The Navy in general looks at electronics much as a woman looks at her automobile. She knows how to run it but hasn't the slightest idea what is under the hood—*and doesn't care.* She likes it big and shiny and with lots of gadgets. If it won't run she takes it to the shop with a grimace and a sigh. If we are going to beat this thing we have all got to understand basically what is under the hood. We have all got to know at least as much about electronics as we do about steam engineering plants, gunnery, and seamanship.

Of course, we could leave it to the specialists as they tried to do with steam in 1836. On this subject the Secretary of the Navy said in his Annual Report of 1864, "Objection may be given that the duties (of line and engineer officer) are dissimilar and that steam-engine driving is a specialty. The duties are not more dissimilar than seamanship and gunnery. When seamanship was the only education given to an officer, it was not believed he could ever teach sailors to drill, and a sergeant of marines performed the duty which is now so admirably discharged by the graduates of the Naval Academy. When gunnery became a specialty it was inconsiderately and unwisely proposed to have a corps of ordnance officers engrafted upon the naval service, a separate or-

ganization, which should draw to itself the knowledge so necessary to each, and therefore proper to be distributed among all officers.

"Fortunately, our naval officers are taught seamanship, gunnery, and the infantry drill, and the service thereby saved from distinct organizations in these respects which would, inevitably, have impaired its efficiency. . . . With the adoption of the suggestions here made, we shall, in due time, have a homogeneous corps of officers, who will be masters of the motive power of their ships in the future as they have been of seamanship in the past."

Again in the Secretary of the Navy's Annual Report of 1865 we find the following advice: "The management of a man-of-war in a gale, on a lee shore, in a narrow harbor, or the splendid manoeuvres of battle by sailing vessels have hitherto been the highest and proudest duties of a thorough naval officer. The skillful disposition of the sails, which was the result of the best training of the old school, is no longer necessary, except as auxiliary to the new motive power which modern invention and science have introduced. The naval vessel is no longer dependent upon the winds, nor is she at the mercy of currents; but the motive power which propels and controls her movements is subject to the mind and will of her commander, provided he is master of his profession in the future as he has been in the past. To retain the prominence which skill and education gave him when seamanship was the most important accomplishment, the line officer must be qualified to guide and control this new element or power. *Unless he has these qualities, he will be dependent upon the knowledge and skill of him who manipulates and directs the engine.*" (Italics supplied.)

The line officer might as well learn electronics. Already electronics has invaded nearly every field in which the naval officer must be proficient. Electronics now aims guns; it drops bombs; it navigates ships; it helps control engineering plants; and it intrudes itself into the sailor's life at every turn. But one thing it cannot do itself is to provide fully qualified electronic repair personnel.

Our personnel situation in electronics was not good last year, and that is an understatement. On board ships the number of regular Navy electronic technicans had increased since 1947, but was still below authorized complement as to both numbers and rates—the Korean War prevented discharges and brought in reservists so that a great improvement was made by the year end. It was the practice to count any striker, or seaman designated as an apprentice for a given rating, as a rated man for the purpose of personnel strength computations. Since electronic technician strikers were almost all graduates of the "class A" or basic electronic schools, there was some argument for this policy. But the inexperienced seaman graduate was not in fact the equivalent of a seasoned third class petty officer, and personnel figures to that extent were fallacious. The veteran of a year on board has been known to say, "It took me six months on board after I arrived from school before I found all of the fuses!"

Not only were units of the fleet below complement but the training of the individual electronic technician was not adequate to meet the demands upon him. Class A school was only basic and there was frequently too desperate a need for his services for him to be sent to advanced "class B" school. The training received at advanced schools, and even at basic schools, was too often more valuable to the individual than his naval career, with the rapidly expanding civilian electronic industry offering attractive jobs. Until the middle of 1950 the training cycle was broken again and

again by the loss of the most experienced men. Those who remained had to carry a double load. One crisis after another came as each aging equipment in its turn broke down. Needed repairs were urgently demanded by captains who had operational commitments to meet. Thus came longer hours and more losses to civilian life of disgruntled technicians. Pity the desperate electronic technician whose basic training is not adequate to promptly cure a complex disorder in a unit of a vital equipment, and whose frustrated officers angrily demand that it be fixed immediately. Working all night he may stumble across the answer. The answer may be to get out of the Navy.

The officer situation was a little better. Sprinkled thinly in the junior grades throughout the service was an increasing number of graduates of the six-month basic officers' course at Treasure Island, California, of the year-long warrant officers' course, and of the wartime schools. Postgraduate instruction in electronics had been received by a comparative few, but the number was increasing. Nearly every ship or unit of the fleet of any size had at least one officer who had received special training in electronics.

The majority of these officers had received basic training with a lot of theory designed to fit them for the administration of electronic installations but not for detailed repair work. A considerable gap therefore existed between their training and that of their average electronic technician. This gap could be closed only by advanced training in maintenance on the part of one or the other. Custom and usage often widened the hiatus. The officer had been trained to exercise direction and not to be a physical doer except in emergency. His duty was to organize, train, lead, and not to lose lightly his capability of overall direction by concentration with his person on the individual task. So the officer might survey the scene, make a rough diagnosis, prescribe a procedure, and then leave his men to struggle, coming back to check on progress and offer encouragement and new directions when needed. The only trouble is that unless one stays with a complex repair job on an electronic equipment continuously he tends to lose the picture, particularly if his own electronic training is basic with a lot of theory. Then, too, the smaller the unit the more watches and other duties and responsibilities make it unlikely, except in emergency, that the officer can personally assume charge of repair work no matter how much he would like to do so.

Both officers and men engaged in maintenance more often than not lacked familiarity with the specific equipment with which they were charged. The multitude of models of electronic equipment made the teaching of all equipments in detail during schooling impossible. "Class C" schools, which teach specific equipments, were attended by a select few. Therefore, a considerable break-in period when the novice became familiar with the circuitry and characteristics of his equipment was the general rule. His instruction book, while an essential tool, was usually no effective substitute for experience. New equipment, which not infrequently was received, required a new period of study and experience to add to the normal workload. The new equipments became more complicated and difficult to repair as time passed until in 1950 informed sources in the Bureau of Ships joked that "each new radar will soon come complete with a physicist to maintain it."

In an attempt to close the gap between the theoretical background of the trained officers and the basic background of the electronic technicians, the Bureau of Ships provided the fleet with the services of numerous civilian field engineers obtained on contract from the various major electronic manufacturers.

These civilian engineers were a great help. Yet conditions did not sufficiently improve with their coming. The training of the Navy technicians to help themselves was a major part of their mission, but as time passed they became more and more essential. Actual labor on deranged equipment used up much valuable time which should have been spent on instruction. The filling out of numerous reports and work sheets for their own companies often cut a half of a working day from their week. From a temporary expedient the contract field engineer was passing into the permanent living proof that the Navy had lost the ability to keep abreast of technical maintenance—that ability which had so long been its pride.

Responsible commanders in the fleet were rightfully concerned about the unsatisfactory electronic performance. They were handicapped severely, however, in their attempts to do anything effective by their own lack of knowledge of the new science and by the lack of experienced senior staff officers who were competent to advise them. For example, a typical type commander's staff had a lieutenant assigned as Electronics Officer. He served as an assistant to a senior commander who was an engineering duty designee. On the squadron commander level the duties of electronic officer were performed by a lieutenant (junior grade) who was the assistant to the staff engineering officer, a lieutenant commander with postgraduate engineering training. All squadron electronics in this particular type command, save one, were also communication officers, an exacting and time-consuming job. While these officers were conscientious in their efforts, their collateral duties cut heavily into their time. Vastly more important, their quite junior status and youth made their voices weak in the councils; their lack of experience devalued their wisdom.

At higher levels this lack of experienced officers persisted. On fleet staffs the electronics officer was a commander. The Bureau of Ships had recognized electronics with the creation of the Electronics Division headed by a senior captain, but few, if any, flag officers in the service in 1950 were well grounded in the field. Navy electronics lacked "horsepower." It lacked the emphasis and direction that officers experienced in electronics and senior enough at all levels could give it.

The materiel condition of Navy electronics at midcentury was not due to personnel deficiencies alone. The equipments themselves were no small factor. World War II made tremendous and overwhelming demands on the designers and manufacturers. Electronic devices gave our forces such an edge in battle that they quickly became an essential for victory. It was no time to indulge in a search for perfection.

In the design of much wartime electronic equipment the manufacturer in effect wrote the specifications as he went along. Circuits were designed using, in the main, components then existing and used in contemporary civilian production. Every new vacuum tube and resistor which had to be specially designed meant the loss of precious time. Is it hard to understand, therefore, how a component which would have an adequate life in a civilian radio, or which for a short wartime career would not fail, could be inadequate in peacetime? Here is the secret of much of our difficulty. Our equipments have grown old and tired. In a piece of sonar gear having a thousand individual components time and use have developed inevitable, serious, and recurring failures which mock the technician at every turn.

There was no time in war to fully balance the conflicting needs of the repairman and the manufacturer. The sets produced were usually much harder to "trouble-shoot" than they could have been. You just cannot get at the gear to fix it without much grunting and

groaning—and wasted hours. The demands for compactness from the fleet were often met at the expense of maintenance facility. There is no intent to criticize these able designers for their wartime work. As it was they worked miracles within the meager allowance of time. The fact remains that today we are still paying a heavy price for that lack of time.

The effect of the regular shipyard overhauls in the maintenance of shipboard electronic equipment was by no means completely salutary. The frequent complaint had been heard, "We went to the shipyard with all of our equipment working, but it took us three months after leaving to get all of the 'bugs' out that the yard put in." None are more acutely aware of this dissatisfaction than the shipyard electronic personnel. They know their business well. But in 1950 the full "system check-out" so dear to the heart of the shipyard electronic shop was not possible. There was no opportunity to get everything electronic, and the things electronics tied into, all ready to test at once—no time to start it all running and keep it going for four or five days *to try to make it break down* and thus eliminate future trouble. Unquestionably, electronic equipment will have "bugs" in it after it has been overhauled. With so many soldered connections, so many individual items which may be about to fail, and so many latent troubles which do not develop until after a considerable period of operation at full operating temperatures and loads, it is understandable. An electronic equipment is not like a pump. A pump can be overhauled and given a full load operating test. It works or it doesn't, and you usually know quite soon. But you cannot say that because you can turn on a newly overhauled radar and get a fine "pip" from a passing plane today, that you will have one day after tomorrow.

Scheduling of work during overhauls can be blamed for most of the trouble. Electronics was still definitely a junior partner in the shipyard firm. In a small space like a submarine control room many trades have to work. Machinists, electricians, pipefitters, and painters, to mention a few, must do major work there. Located there is the air search radar, radio equipment, fathometer, and other electronic equipment. The electronic engineers were nearly always pushed to the last or given the odd moments. In the midst of the last-minute tests the painters, like as not, arrived to mask everything and flourish their spray guns; or the electricians had not finished the cabling and no suitable power was available. Not until the final few days of the overhaul were all components of the major electronic systems, including fire control instruments receiving electronic inputs, ready for test as a whole.

Rarely during the test runs at the end of an overhaul were aircraft or submarines available to give the final proof of the radar and sonar equipment. Instead, if the indications were satisfactory it was "signed off" as completed. It had to be. There was no more time.

A grave handicap for those who fought the battles of maintenance was in the supply of spare parts. In the mountains of electronic spares left over from World War II one might believe would be an ample store of parts of every variety. This was not the case. Shortages of some items were critical in 1950 for several reasons.

The complex nature of electronic materiel means the training of a special breed of storekeeper qualified to identify and handle it. A man who cannot deal with transducers, magnetrons, modulation networks, and fusetrons cannot cope with electronic supply. If he tries, as many were forced to do after World War II, a hopeless muddle is likely to ensue. The shortage of skilled storekeepers was a big problem of the elec-

tronic supply system. Shortages developed at many points due to lack of time to identify and ship material from activities who stocked it, or due to mistaken identification and shipment of the wrong item.

Shelf life of some components, notably vacuum tubes, was a potent factor. Some tubes become gassy or otherwise defective after a year or so of nonuse. Large stocks of tubes were found to be defective upon issue. The percentage of bad tubes issued varied widely by no predictable pattern. Some special tubes were about fifty percent defective. Shelf life also affected the service life of many items; old components were likely to fail quickly.

In the educated guessing that had to be done in wartime when deciding upon the number of spare parts to be stocked for a given equipment errors were inevitable. Now we know, on the basis of failure data, that additional numbers of certain items should have been procured. It is hindsight, and it is expensive now for the taxpayer.

Peacetime supply policies also create temporary shortages. Obviously, we cannot afford to maintain huge duplicate stocks of electronic materiel at every fleet base and supply center. Unused stocks of special spare parts become obsolete with the equipments. The less we have on hand at that juncture the better. Also the more money we can divert from the purchase of spare parts to the purchase of new equipments the better. Yet we must have an adequate peacetime and wartime reserve of parts. A fine balance must be struck. The attempt was made to stock parts at a central stock point with other supply activities only carrying enough for immediate requirements. Too often dispatch action was necessary to procure an essential item.

The determination of requirements was a great problem. In spite of repeated strong pleas from the Bureau of Ships for the submission of failure reports on each failure of a component, no matter how small, the response from the field was spotty. The failure report was one of the simplest forms yet devised. It was designed to be filled in in pencil and mailed by the enlisted technician himself directly to the Bureau. It appeared entirely painless, but the electronic personnel of the fleet apparently could not believe that the reports would do any good. They did not understand the reason for a failure report on a tube used in everyday home radio sets. They hated paperwork. Without this means of knowing accurately the rate of failure of components, the Bureau of Ships was severely hamstrung in its efforts to obtain a sufficient number of the necessary components, and to decide when component design needed improvement.

Finances profoundly affected 1950 Navy electronics. Since there was not enough money to have our cake and eat it too, the cash went to research and development. Replacement equipments purchased were few and far between. New pilot models were procured, however, for new construction ships and aircraft. When the Bureau of Ship's eleven millions of dollars for purchase of new electronic equipment in fiscal 1950 is contrasted with the pre-Korean forty-four millions provided for that purpose in fiscal 1951, it is obvious that things are definitely brighter in this area.

If there were serious problems so, too, were there solutions to these problems. At the center of any comprehensive program for improvement should be the recognition by all of the importance of electronics. Most will protest that they do recognize its importance. Many will be right. But before we can expect real improvement in Navy electronics we must open our eyes to the many glaring inconsistencies which still enjoy life and health. Electronics has not been recognized for its

true importance when this indispensable branch of engineering is run by juniors. Like a pesky little sister being ditched by older brothers, electronics gets ditched by the senior shipboard and staff officers. To let electronics go to the junior man who cannot avoid the job has been more often the rule than the exception. To assign heavy collateral duties, such as communications, to the electronics officer is common practice. As long as electronics is represented by the junior man it will be junior in performance and junior in progress.

The standard organization of fleet units and staffs should be overhauled. On the new command ship (CLC) being constructed, should electronics be relegated to the Electronic Repair Officer, a subordinate of the Engineer Officer; or to the Communication and Combat Information Center Officers who are subordinates of the Operations Officer? Electronics has reached the stage on combatant vessels where it merits departmental status on an equal basis with Gunnery and Engineering. It follows that the electronics officer on a staff should have equal rank and status with the gunnery officer and the engineer officer.

Nor has electronics reached full recognition when the emphasis in shipyards is on getting the ship out of the yard in the allotted period with the engineering plant and hull in perfect shape and electronics is given a lower priority. In war a ship that departs from a shipyard with hull, machinery, and ordnance in the best of condition, and yet fails to detect an attacker before it is too late due to defective electronic gear, would have been better off never to have sailed. The loss of a ship away from base usually means total loss of the ship and great loss of life. If the ship had been sunk in port, unable to leave due to engine trouble, probably fewer lives would be lost and salvage might be possible. Not that the hull, machinery, and ordnance equipment is not essential, but rather that the electronic equipment is equally so.

A new overall attack on the electronic personnel situation should be made. One solution which has been suggested is that the complicated electronic equipment be kept healthy like the human body. Let the electronic technician be the "hospitalman." Provide a new officer specialist on every ship who would be the "doctor" and have analogous duties including "surgery" when necessary. This proposal is radical to the naval service. It would disturb many well-founded and proven relationships between the line officer and the specialist, and between officers and men. Nevertheless, new and more baffling equipments are coming before we have even mastered the present ones. Something must be done now to meet the future problems.

A workable compromise might be to promote the bright electronic technicians, after suitable schooling and sea service, to the warrant officer grades at a much accelerated rate. Warrant billets should then be established in the field of electronics in every unit of the fleet of any size. Fully half of the service of these warrant officers should be in advanced schools or shipyards where they could keep up to date on the latest techniques and equipments. The promise of increased opportunity should hold a much larger share of the best electronic technicians in the service in peacetime. Also the line officers would be able to continue their overall command in electronics without the present gap between their experience and training and that of the enlisted technician.

The various Navy electronic schools at present are doing a splendid job of training as the offers by civilian firms prove. Unless the present chain of events can be broken, however, a small number of electronic technicians will be

given advanced training. If we had an inflexible rule that all electronic technicians were to return to advanced school automatically after completion of three years of field service, we could show a positive improvement. Electronics is not static. New schooling is required at regular intervals if the technician is to keep abreast of new devices and methods. Advanced training is essential to advanced "trouble-shooting."

Adequate maintenance depends more upon the solution of the personnel problem than upon any other other excepting only the education of the service at large, from which such a solution would certainly flow. Given sufficient properly trained maintenance personnel many of the present troubles would vanish.

Shipyard overhauls are a tough nut to crack. Elevation of the electronics work to equal priority is not the entire answer. Rearrangement in the involved, highly specialized business of scheduling the work on a vessel in overhaul is not to be undertaken lightly. That is not to say that it cannot or should not be done. It has been done before. At one shipyard not long ago the heat was on. A submarine had left the yard and in ten days all major electronic equipment was out of commission. Loud complaints were received from quarters that could not be ignored, with the usual ominous copies to all concerned. The next submarine work had to be good or else. It was decided that two weeks from the end of the overhaul the control room and conning tower would be closed to all trades except electronics during the day shift—and most of the other trades were directed to be finished in those compartments by that time. In particular, all normal electric power had to be restored by the deadline, and all associated fire control equipment was to be in full operation. As a sample of the ship's attitude, the commanding officer stated that if not one drop of paint was used anywhere except behind equipment where the ship's force could not reach, he would be satisfied—*provided only that everything work when he left and stay working.* The ship would do the painting. Of course, the two-week deadline could not quite be met. But a full ten days was saved for electronics. Most of that ten-day period was spent continuously operating the equipment. Ship's force and yard personnel took turns watching the gear. When it failed the shipyard engineers made repairs as rapidly as possible and turned it back on. Needless to say this program was a big success. To prove optimum operation the services of aircraft and a surface ship were arranged and all equipment was checked out and given final adjustments. No significant casualties to any electronic equipment followed departure from the shipyard, and the submarine was able to give the yard a well-earned pat on the back.

In the field of design much is now being done to keep the future technician from unnecessary lost motion. Here is a vital spot. If we have complex gear that we cannot keep operating, we would be better off without it—so that we could use the space for a few more rubber life rafts.

Components are the bane of the technician's existence today, the lack of them and their short service life. Redesign of components as to size, life, and susceptability to mass production is receiving much attention. These requirements are often conflicting. Specifications must be drawn with the utmost care. Essential failure data must be provided by the operating forces. Rather than the voluntary submission now relied upon, the report should be made rigidly obligatory. It is absolutely necessary that complete data be collected and analyzed. The only way to do it is to rely on command, the military way. The technician will not fill out a form that takes only one second of his time for some distant and vague bureau if

his commanding officer is not concerned with it and wants the equipment fixed. And he will continue to be needed to fix the equipment all of the time until he gets new components. Instead of asking the technician for the information, we should ask the commanding officer. We should clearly inform the commanding officer why the information is needed, and exactly what is being done with it.

Liaison with the operating forces is nowhere more important that in design work. It is difficult to see how there could be too much of it. It benefits both designer and consumer. The consumer learns what the designer has up his sleeve and what it is reasonable for him to ask for. A demand is created for hitherto unknown items. The designer is forced to come down out of the clouds and fit his schemes to the needs of the consumer. He secures direction and inspiration from the man in the field. Those primarily concerned with tactics, and even strategy, must be fully capable of incorporating new developments into their plans and even of demanding them from the designers. A knowledge of what our own designers are working on is essential to those who must wonder what the enemy's bag of tricks contains, and how to counter them.

If two world wars reemphasized any lesson, it was that of supply. Supply is closely tied to demand. Design must help supply to meet the demand. The demand in peacetime, if one can call this peacetime, now frequently outstrips the supply. Economy of force in the utilization of the available funds requires that we juggle supply and demand so that we do not tie up more money than is necessary in idle parts. We must remember that successful war is the responsibility of the military man; we are not preparing for peace. We must try not to let peacetime methods and ideas lose our campaigns.

The United States Navy has many achievements in electronics of which it may be proud. We are today ahead of every other nation in our applications of electronics to naval use. It would be folly to rest now, and we will not do so. What vast new fields electronics will open to us in the next hundred years none can predict with certainty. If the naval officer of 2050 does not, however, look upon many of us with the same indulgence with which we now regard the sailpower captain of 1850, it will be surprising. That we could fail to appreciate the great tools we have and fail to strive with all of our energy to master them, is evidence of the inherent conservatism of the human being. That we are doing far better with our new science than our forebearers were doing a century ago may indicate that man is learning something after all.

A great educational job, even a missionary job, remains to be done within the naval service. The aim must be to place electronics on the level of importance in the line officer's heart and mind with time-honored seamanship, ordnance, and engineering. Within the memory of living man this missionary work was being conducted for engineering. Today the engineer is not the object of sly mirth or of the wise tolerance of the ignorant. The captain of a man-of-war of 1850 would be amazed to find the once lowly engineer envied and respected in 1950. History will repeat itself.

THERMAL BOLT STRESS STUDY IN A HIGH-PRESSURE FLANGED PIPE JOINT

J. H. POTTER
AND
W. W. EITEL

THE AUTHORS

JAMES H. POTTER was born in New York, N. Y., in 1912. He attended the public schools of that city and later studied mechanical engineering at Columbia University. Following the M.E. degree in 1936, he worked for four years in the chemical and shipbuilding industries. He was on the engineering faculty at the Johns Hopkins University for nine years, serving as Instructor (1940), Assistant Professor (1943), Assistant Dean of Engineering (1946). He is presently Professor of Mechanical Engineering at the University of Illinois, a post he has held since 1949. He received the Dr.Eng. degree at Johns Hopkins in 1948.

While employed in the Federal Shipbuilding & Dry Dock Company, he became interested in the problem of transient thermal stresses in the bolts of high pressure flanged joints. Actual investigation, however, had to be postponed until university laboratory facilities were available.

He has been a member of the A.S.N.E. since 1939.

W. W. EITEL was born on October 15, 1922, in Baltimore, Maryland. He is a graduate of the Johns Hopkins University. Upon graduation he continued his education as a part-time graduate student and from 1946 to 1950 as a full-time graduate student in the Mechanical Engineering Department of the Johns Hopkins University. He is presently employed as a project engineer for the Westinghouse Electric Corporation in Baltimore, Maryland.

SUMMARY

This paper reports on a series of temperature runs on a high-pressure, high-temperature bolted flanged joint. The work was done in the mechanical engineering laboratory at the Johns Hopkins University during 1949. Bolt stresses were computed from temperature measurements in the bolts and flanges. A stream of hot gases was introduced in the line containing the flange, the stream temperature ranging from 600°F to 1200°F. The Reynold's Number varied from 10,000 to 20,000.

The temperature differential between the flange and bolts was found to be small in all cases, and linearly related to the stream temperature. The findings are contrasted with those reported earlier by some German engineers.

NOMENCLATURE

The principal quantities used in this paper have the following definitions and units:

T_s = stream temperature, °F.
T_m = metal temperature, °F., location as noted in application.
T_F = flange temperature, °F., at some point near a bolt hole.
T_f = average flange temperature, °F.
T_B = specific bolt temperature, °F.
T_b = average bolt temperature, °F.
E = Young's modulus, psi.
α = mean coefficient of linear thermal expansion, inches per inch per °F.
ζ = "Thermal stress" specifically used to identify the stress due to the flange-bolt temperature differential.
Re = Reynold's Number, dimensionless.

I—INTRODUCTION

Widespread interest in the analysis of the flanged pipe joint has extended over a great many years. An analogous treatment of rigid members joined by an elastic bolt and gasket was presented as long ago as 1897.[1] The problem of selecting and applying bolt loads has been described by Petrie[2] and others, while flange stresses have been the object of many investigations.[3] In an effort to operate with lower bolt stresses, hairpin-type springs were installed between the flange and one nut on an experimental joint used at one of the power stations of the Detroit Edison Company.[4] This technique limited the bolt stress to about 10,000 pounds per square inch, and the test installation functioned satisfactorily on high-temperature steam service.

The danger of using high initial bolt stresses in a joint intended for high-temperature service was pointed out by Salingrè,[5] who offered a nomographic device for estimating the stress increment due to thermal expansion. The Salingrè article also mentioned temperature studies made by A. Borsig of Berlin on a bolted flanged joint in a power house. The Borsig tests were made on a line supplying steam to an engine, and report time-temperature relations for a joint with and without insulation. The Borsig data are shown plotted in Figs. 1 and 2.

During the warming-up period the pressure as well as the temperature rose. This fact, the odd shape of the test joint, and the absence of data on flow conditions and physical dimensions, make the Borsig tests indicative rather than definitive. They do show a large temperature difference between the bolt and the flange, with the implication that a considerable thermal stress has been added to the bolts. In another

part of his article he indicates that these thermal stress increments may exceed the original cold pull-up stress.

The present investigation was undertaken in an effort to determine the magnitude of the thermal stresses in the bolts of a high-pressure, high-temperature flanged joint using a gaseous medium. In order to approximate extreme conditions, a hot gas stream was suddenly introduced into the line containing the joint, so that transient stresses might be investigated. To obtain a maximum temperature differential between the flange and the bolts the joint was left uninsulated.

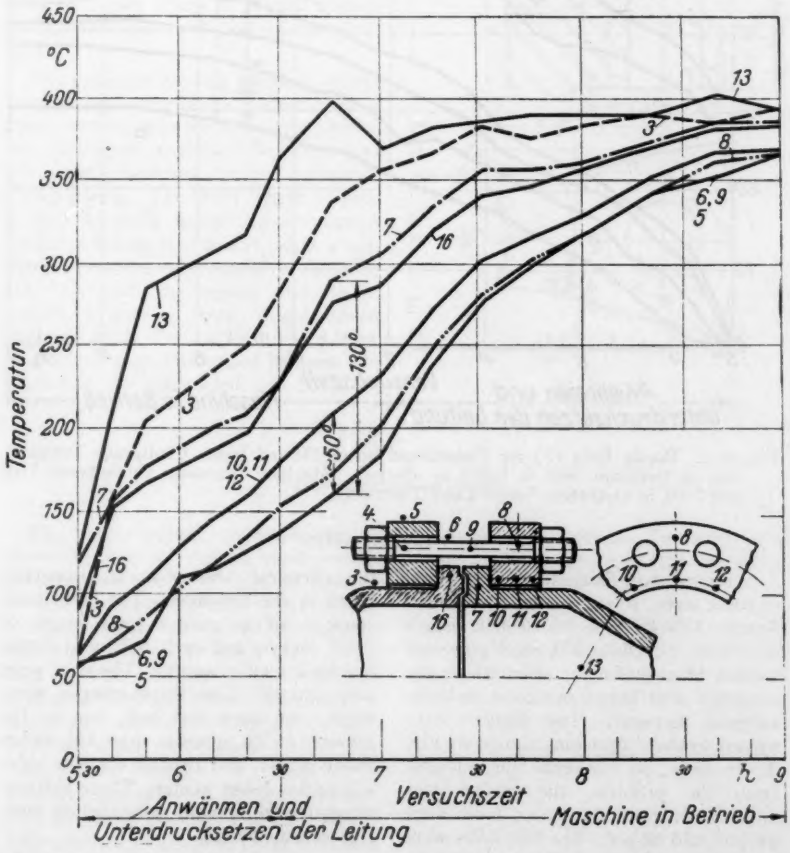


FIGURE 1. Borsig Data (5) for Insulated Joint Heated by Steam. Ordinate shows Temperature, °C., abscissa the time in hours. Machine warming up from 5:30 to 6:30, and in operation under load thereafter.

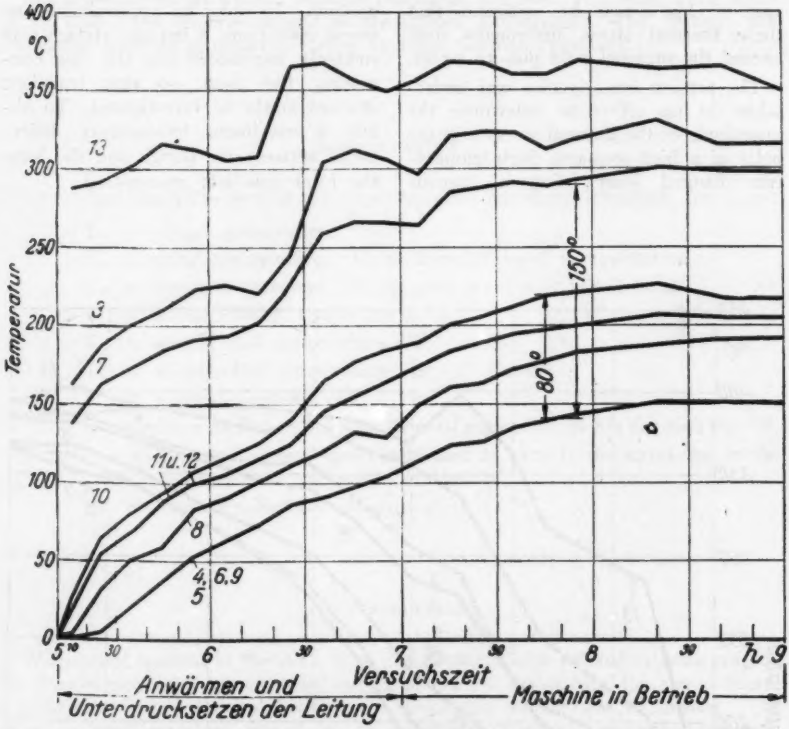


FIGURE 2. Borsig Data (5) for Uninsulated Steam-Heated Joint. Centigrade Temperature as Ordinate, time in hours as abscissa. Machine Warming Up between 5:10 and 7:00, in Operation Under Load Thereafter.

II—THE TEST JOINT

A forged-steel flanged joint was used in these tests. It was of the commercial 4-inch, 1500-lb. type. A 6-inch length of 4-inch, Schedule 160, steel pipe was welded to each flange, after which the assembly was bored out to a uniform internal diameter. The flanges were turned to the dimensions shown in Fig. 3. In order to eliminate one variable from the problem, the gasket was omitted. Instead, the raised faces were ground and lapped. The bolt holes were faced so as to offer a flat bearing area for the stud nuts.

Eight 1¼" bolts were used, having the chemical and physical properties listed in the appendix. The bolts were squared off to give a total length of 7.975 inches, and each bolt was identified by a number punch. The nuts were also squared. Five thermocouples were located on each test bolt, one at the center, two on opposite sides 1⅜ inches above center, and two on opposite sides 1⅜ inches below center. These thermocouples were actually inbedded in inactive screw threads.

Thermocouples were located in the flanges as shown by the lettered positions in Fig. 3. In the preliminary tests,

28ga. iron-constantan thermocouples were used. These had been calibrated previously against the boiling points of water, sulphur, and mercury. In the later tests chromel-alumel thermocouples were used, and recording potentiometers were employed. The thermocouples in the flanges were protected by capillary glass tubing.

During the assembly a special fixture was used to center the bolts in the bolt holes. The bolts were carefully tightened to give a root stress of 50,000 pounds per square inch or 44.9% of the yield point.

Any external bending moment on the flanged joint would be reflected in a stress increment in the bolts. To remove this additional variable, the joint was not rigidly fastened to the communicating piping. The short length of pipe welded to each flange merely butted against a 4-inch steel line through which the hot products of combustion could be led. To hold the piping and flange axially, a narrow strap of galvanized iron was used. For gas tightness a layer of asbestos paper was used between the pipes and the galvanized iron band.

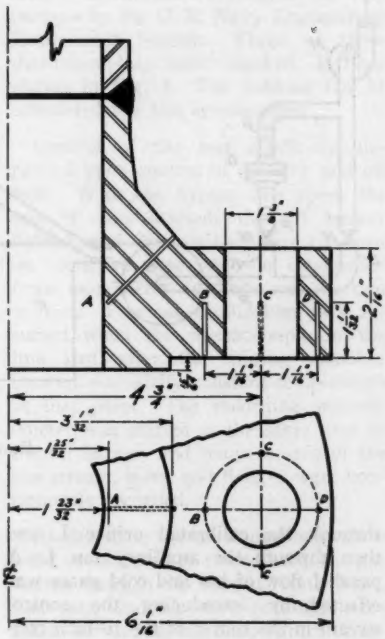


FIGURE 3.

III—METHOD OF TEST

The experimental arrangement is shown in Fig. 4. Air at room conditions entered the 3-inch pipe A, passing around the control cones. After a long straight run, the air passed through the thin plate orifice B. Both air orifices used in this research were calibrated against a Thomas Meter. A precision manometer, which could be read to 0.001 inch of water, was used during the tests. The chamber C contained a domestic gun-type oil burner (Anchor Post Products Co., type P-3C) equipped with several nozzle plates and having a maximum capacity of three gallons per hour. Oil was stored in the 50-gallon tank D, which also supplied fuel during the warming-up period. The scales and weigh tank E were used for weighing the fuel burned.

The combustion chamber F consisted of a steel drum lined with 4 inches of plastic firebrick. A 3-inch layer of glass wool was used externally. The hot gases passed from the combustion chamber through a ten-inch line to the mixing chamber G, entering the latter tangentially.

In order to control the temperature of the gaseous mixture, a cool air system was added. Air at room temperature entered a 6-inch line at H, passed

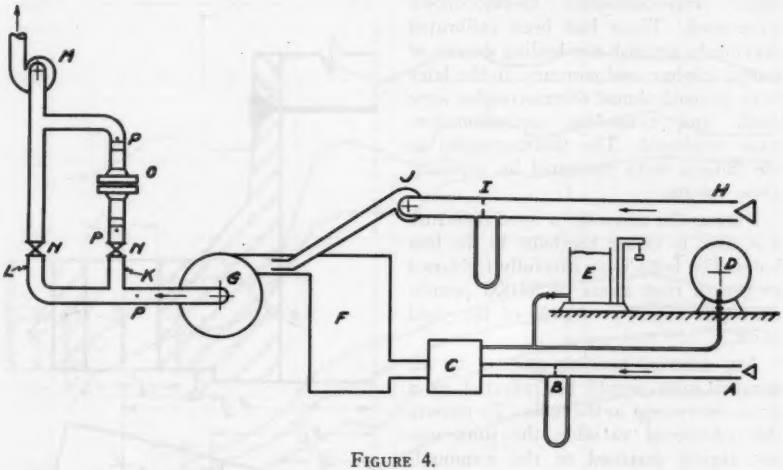


FIGURE 4.

through the calibrated orifice I, and then through the auxiliary fan J. A parallel flow of hot and cold gases was effected by introducing the control stream in the center of the 10-inch connecting line. The offtake from G was centrally located in an effort to promote the mixing of the two streams.

From the tank G the hot gases were led into a 4-inch line. The gases could be sent through the test line K or through the bypass line L. Disposition of the fluid stream was controlled by the quick-opening valves N,N, one of which was always open. These valves were constructed for this experiment, and each was actually a special type of "spectacle" flange. The moving members were large triangular pieces of half-inch transite board, covered top and bottom with asbestos paper. The valve moved about a fulcrum located at the apex of the triangle. Special clamping frames were located on the communicating piping above and below the transite plate. When opening the valve the clamps were reversed, lifting the pipe ends free of the plate. The moving element was then shifted, and the clamps could then be brought down to force the pipe ends against the asbestos paper, making a gas-tight joint. One of the quick-opening valves is shown in Fig. 5. The valve with its clamping frames is shown just under the tripod which supports the flange. Throughout the test program these inexpensive valves operated most satisfactorily, being gas tight and positive in action.

Returning to Fig. 4 it will be seen that the gas path above the quick-opening valves was either through the bypass line L, or through the test section. The flanged joint was supported independently of the main piping as noted previously. This support was provided by the pipe tripod shown in Fig. 5. Small stainless steel cones were centered on each of three bolts, the cones being supported on short lengths of ½-inch quartz tubing held on adjusting screws. The joint was braced laterally by three guy wires attached to the upper section of the assembly.

Stream temperatures were measured by calibrated silver-shielded chromel-

FIGURE 5. Strain Cycle Analyzer and Gage.

alumel thermocouples, loaned for this purpose by the U. S. Navy Engineering Experiment Station. Three of these thermocouples, each marked P, are shown in Fig. 4. The exhaust fan M completed the test arrangement.

Control of the test conditions depended upon control of air flow and oil flow. With the bypass line open, the fan M was started, the oil burner lighted, and the auxiliary fan J turned on. A given flow rate was established from manometer readings on the two orifices. Thermal equilibrium was assumed when the thermocouple in the line just after the mixing chamber showed essentially constant temperature at that point. The recording potentiometer was started at the same time as the oil burner, and temperatures of the gas stream, bolts, and flanges were continuously recorded.

At this time the quick-opening valves were reversed so as to send the hot gas stream through the flanged joint. Time, fuel rate, and air flow readings were taken, as well as the temperature measurements.

IV—RESULTS AND DISCUSSION

A schematic representation of the temperature history of the joint is shown in Fig. 6. The oil burner was turned on at Point A, the products of combustion being by-passed to atmosphere. The stream temperature, T_s, showed a rapid increase.

Due largely to conduction, there was a gradual rise of temperature at the flanged joint even though it was blanked off. When stream conditions were stabilized enough for test purposes, the flange temperature had risen to point D, and about five and a half hours had elapsed.

Immediately after the change-over, a continuous increase in the temperatures

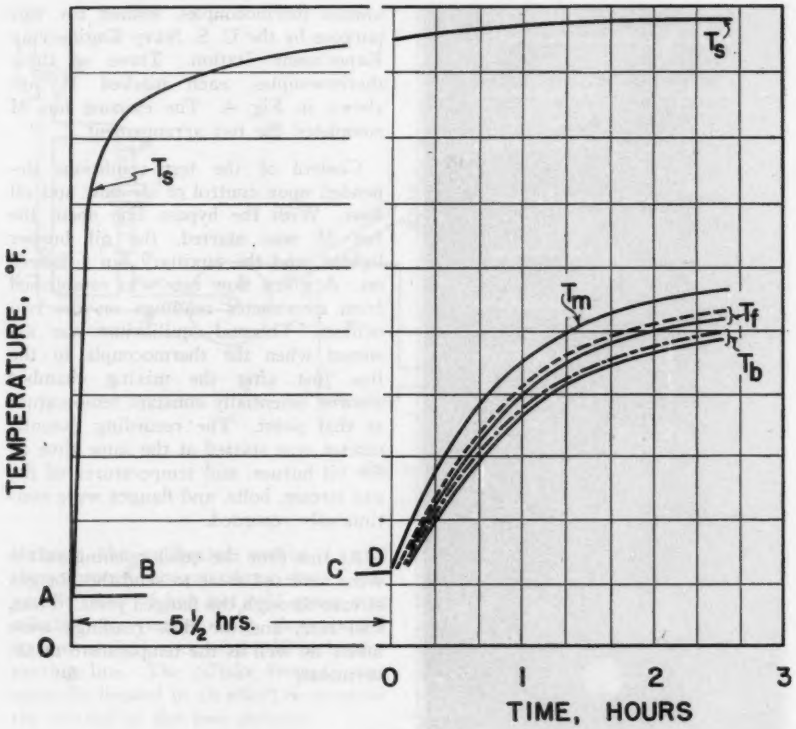


FIGURE 6.

on the flange and bolts was measured, each tending asymptotically toward a constant value. Referring to Fig. 3, the flange metal temperature was measured at point A, this thermocouple plots as T_m in Fig. 6. Below this there is a narrow band representing the range of flange temperatures T_F. The lowest group of curves shows the limits of the bolt temperatures T_B.

While the flange and bolt temperatures increased continuously, the differential temperature between these two tended to reach a constant value in a relatively short length of time. This is shown in Figs. 7 and 8, where the difference between the average flange temperatures and average bolt temperatures in shown plotted against time. In these graphs the large numerals in circles represent run numbers. In general, the maximum flange-bolt temperature differential was established somewhere between 20 and 40 minutes after starting. The only exception was Run No. 11, the maximum stream temperature test, which required 75 minutes to level off. The significant quantities on the several runs are summarized in Table I.

The equilibrium, or maximum, value

TABLE I

Run Number	1	2	3	4	5	6	7	8	9	10	11
(Tf−Tb)eq.	20.0	16.5	25.0	29.0	25.0	30.0	25.5	27.5	32.0	29.5	37.0
Time to eq. (minutes)	47	44	40	42	36	32	28	30	28	33	75
(T_s, (°F)	799	608	920	955	857	960	922	875	981	1071	1233
Reynolds No. (x10-3)	14.95	13.14	20.70	12.46	18.20	11.82	17.70	12.50	17.43	10.13	10.92

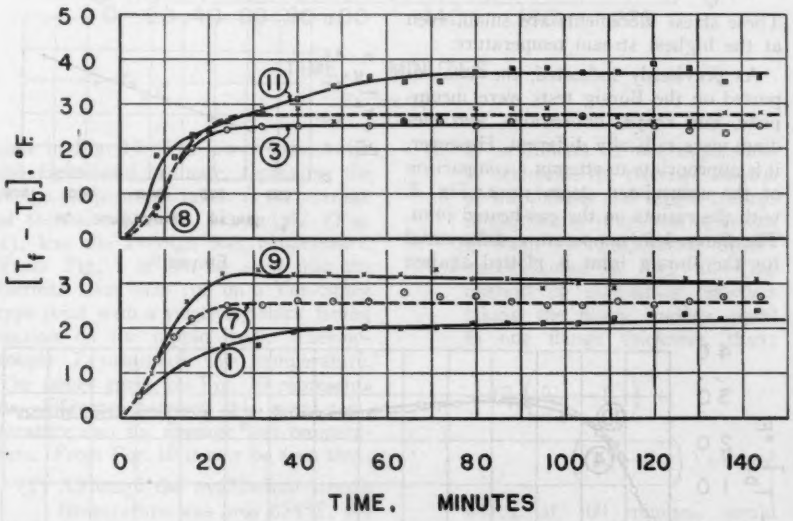


FIGURE 7.

of $(T_f - T_b)_{eq.}$ is significant as an indication of the additional bolt stress imposed by differential thermal expansion. In Fig. 9 these values have been plotted against stream temperature. It is interesting to note that this temperature differential is not large, and that there is an essentially linear relation between it and the stream temperature.

Over the whole range of the tests the greatest flange temperature was 524°F, and the lowest bolt temperature was 279°F. In this range, and for the materials involved, an average value of the coefficient of thermal expansion of 7.58×10^{-6} inches per inch per degree F. may be used. Taking a mean value of the modulus of elasticity of 28.3×10^6 pounds per square inch, the thermal bolt stress is given by:

$$\zeta = \alpha E \left[T_f - T_b \right]_{eq.} = (7.58)\ (28.3)\ (T_f - T_b)_{eq.}$$

Equilibrium values of the thermal stresses are given in Table II.

TABLE II

Run Number	1	2	3	4	5	6	7	8	9	10	11
ζ, psi	4280	3540	5360	6220	5360	6430	5460	5890	6860	6320	7940

These stress increments are small, even at the highest stream temperature.

As previously indicated, the data reported on the Borsig tests were incomplete, and shape, dimensions, and medium were radically different. However, it is appropriate to attempt a comparison of the uninsulated steam joint, Fig. 2, with the results on the gas-heated joint. The flange-bolt temperature differential for the Borsig joint is plotted against

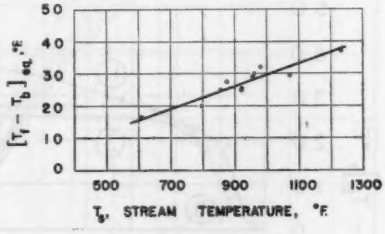


FIGURE 9.

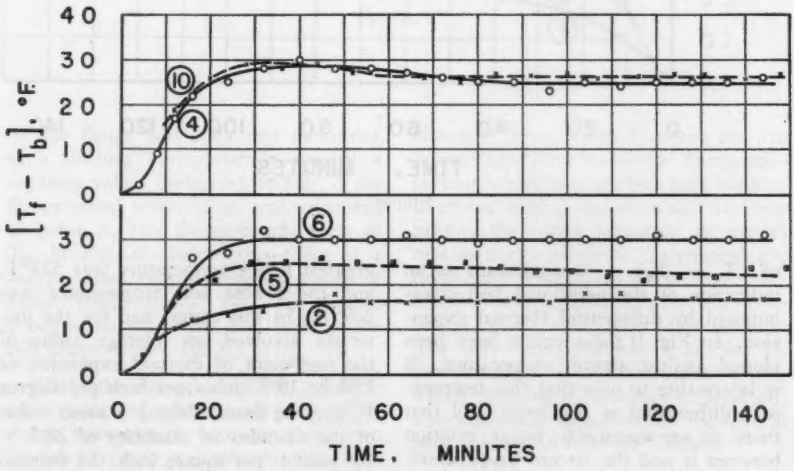


FIGURE 8.

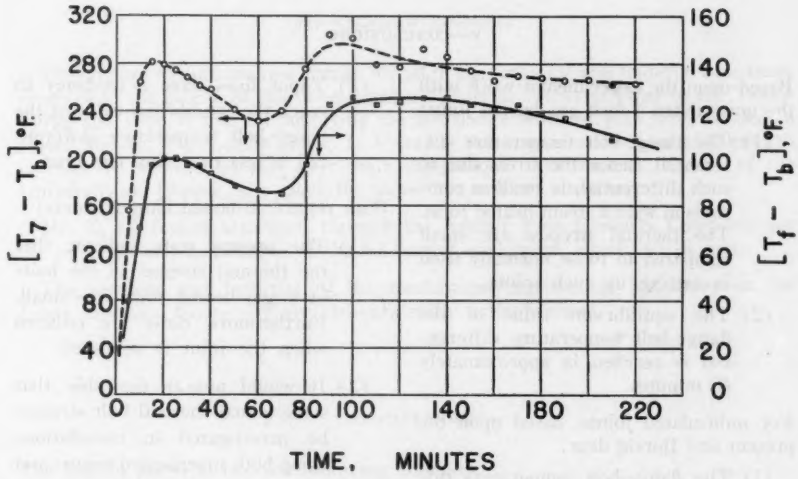


FIGURE 10.

time in Fig. 10. The lower curve, with the right-hand ordinate, represents the flange temperature, taken as the average of thermocouples 10, 11, and 12 (Fig. 2), less the average bolt temperature. From Fig. 1 it can be seen that the German tests were run on a Van-Stone type joint with a relatively thick facing section on the nipple itself. Thermocouple 7 measured this temperature. The upper curve on Fig. 10 represents the difference between this metal temperature and the average bolt temperature. From Fig. 10 it may be seen that:

(1) Although the equilibrium stream temperature was only 654°F, the flange-bolt temperature differenitals were very much greater than for the gas-heated joint with a stream temperature of 1233°F.

(2) The wavy form of the curves suggests that the temperature and/or steam pressure were varying. If this is allowed for, the flange-bolt temperature differential is seen to build up toward an equilibrium value at a rate faster than that shown in the gas-heated joint.

(3) It is difficult, if not impossible, to determine the effective lengths of the flanges and nipple facings in the Borsig joint, and to assign to them the temperature differences shown in Fig. 10. One method of estimating involves taking the nipple facings equal to one flange thickness, then:

$$\frac{1}{3}\left(2\left(T_f - T_b\right) + \left(T_7 - T_b\right)\right)\alpha E = \zeta$$

which, at 100 minutes, would have a value of:

$$\frac{1}{3}\left(2\left(124\right) + \left(296\right)\right) \times 7.58 \times 28.3 = 38{,}800 \text{ psi.}$$

Stresses of this order are anticipated in the Salingrè nomograph. The high flange-bolt temperature differentials, and consequently the high thermal-bolt stresses, in the steam joints are apparently due to the very high film coefficients of heat transfer on the steam side.

V—CONCLUSIONS

Based upon the experimental work with the uninsulated 4-inch gas-heated joint:

(1) The flange-bolt temperature differential, hence the stress due to such differential, is small in comparison with a steam-heated joint. The thermal stresses are small compared to those normally used in setting up such joints.

(2) The equilibrium value of the flange-bolt temperature differential is reached in approximately 40 minutes.

For uninsulated joints, based upon the present and Borsig data:

(1) The flange-bolt temperature differential appears to build up steadily to an equilibrium value, and does not pass through any transient values in excess of that at equilibrium.

(2) Vapor lines have a tendency to reach the equilibrium value of the flange-bolt temperature differential in less time than gas lines.

With regard to bolted flanged joints:

(1) The present tests indicate that the thermal stresses in the bolts of a gas-heated joint are small. Furthermore, these are reduced when the joint is insulated.

(2) It would appear desirable that some actual thermal bolt stresses be investigated in installations using both superheated vapors and condensing vapors, as the present research has not touched upon these problems. The Borsig tests have indicated a high magnitude for such stresses.

VI—BIBLIOGRAPHY

(1) THE LOCOMOTIVE, Vol. XVIII, No. 11, Nov. 1897, pg. 161. Hartford Steam Boiler Company.

(2) "Methods of Applying and Determining Bolt Stresses"—E. C. Petrie, HEATING, PIPING, and AIR CONDITIONING, Vol. 8, June, 1936, pg. 303.

(3) "Design of Bolted Flanged Connections"—Waters, Wesstrom, and Williams, MECHANICAL ENGINEERING, 1934, pg. 736.

(4) "High Temperature Steam Experience at Detroit"—P. W. Thompson and R. M. Van Duzer, ENGINEERING (London), Dec. 22, 1933.

(5) "Machinenteile für Hochdruckheissdampf"—Salingrè, z.V.D.I., Vol 74, part II, Sept. 6, 1930, pg. 1237.

VII—ACKNOWLEDGMENT

The writers wish to express thanks for all of the technical and material assistance received in the course of this investigation. Especial thanks are due to the Taylor Forge and Pipe Works for supplying the test flange with the pipe leads welded in place. Messrs. J. E. Lattan and P. L. Vissat of that organization were kind enough to read the manuscript and offer their suggestions. Professor S. Konzo, of the University of Illinois, also gave the paper his critical attention.

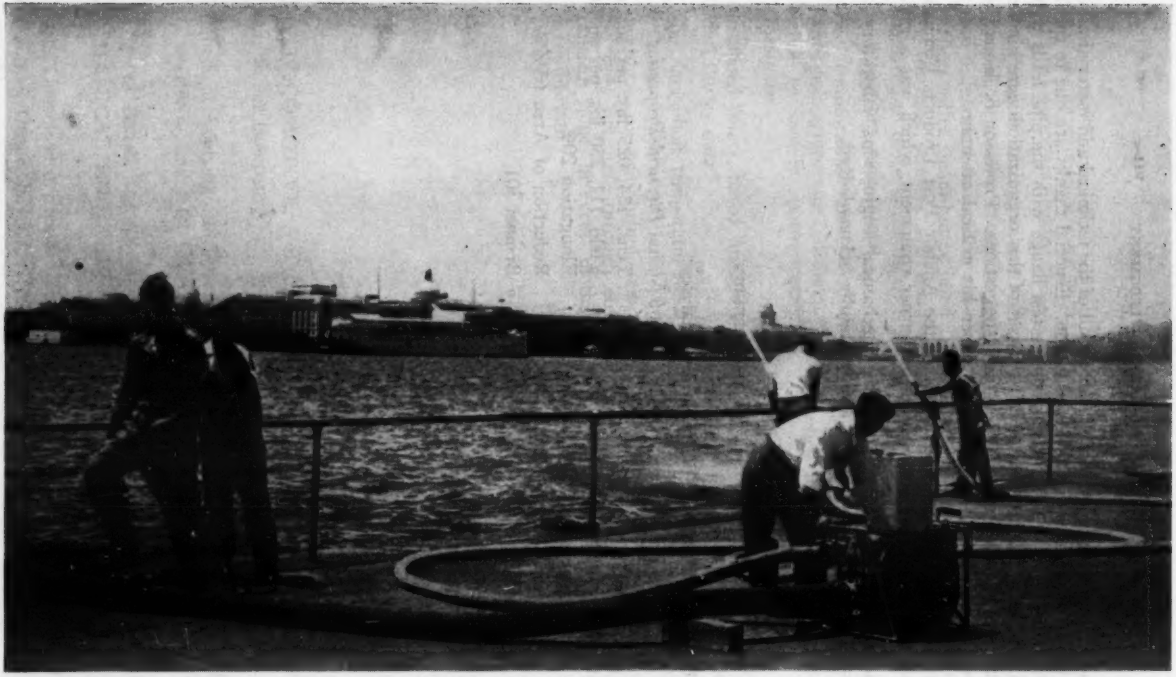
Mr. T. H. Smoot arranged, through the Anchor Post Products Company, for the selection and installation of the oil burner equipment, which was a gift.

The services and facilities of the mechanical engineering laboratory at the Johns Hopkins University are also gratefully acknowledged.

VIII—APPENDIX

The bolting material had the following analyses:

Chemical Analysis		*Physical Properties*
Carbon	.40	Tens. 134, 025 lbs. per sq. in.
Manganese	.69	Yield 111, 250 lbs. per sq. in.
Phosphorus	.021	Elongation 20%
Sulphur	.027	Reduction of Area 63½%
Silicon	.21	Brinnel 301
Chromium	.96	
Molybdenum	.18	



Portable Gas Turbine Pump In Use

THE T-45 POWER PLANT

A SMALL GAS TURBINE PRIME MOVER DRIVING A PORTABLE FIRE PUMP

A. D. ZAKARIAN AND ROY R. PETERSON

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ROY R. PETERSON received an ME degree from Stevens Institute of Technology in 1939. From 1939 to 1945 he was employed by the Atlantic Refining Co. in the Engineering and Construction Dept. He served in the Navy from 1945 to 1947. Since release to inactive duty, he has been active in the gas turbine development program of the Bureau of Ships.

GENERAL

With the advent of the gas turbine engine as a prime mover, it becomes increasingly evident that units of low power output can be used to advantage in applications formerly considered beyond their scope. Where the inherent advantages of the gas turbine engine outweigh the moderate thermal efficiency loss attendant on scale reduction, a small gas turbine can be applied with major gains. A simple cycle turbine scaled down to an output as low as 50 shaft horsepower still gives a weight-to-power ratio considerably better than a comparable reciprocating gasoline engine. It can operate on low-cost and less flammable Diesel fuel, and has potential reliable starting characteristics at low ambient temperature. There are fewer parts in a gas turbine

than in a reciprocating engine, and hence the maintenance and serviceability are correspondingly more simple. These facts, especially the weight-power ratio, led the Navy Department to consider such a power plant to drive a 500-gpm. portable fire pump for shipboard use in place of the pumps now employed, which are driven by four-cylinder, two-cycle gasoline engines.

By 1948 the intensive development and improvements in gas turbine design and the widespread service experience gained in aircraft applications indicated that an acceptable operating life could be expected from the small gas turbine power plant under consideration. Consequently, competitive proposals were requested by the Navy, and contracts were awarded by the Bureau of Ships for prototype units for test. One of the contracts was given to Solar Aircraft Company of San Diego, California, which designed and produced a power plant identified as the Solar Model T-45 Portable Gas Turbine Driven Pump. The prototype pump set was delivered in June 1950 for evaluation tests at the Naval Engineering Experiment Station, Annapolis, Maryland. It is the purpose of this paper to discuss the mechanical construction, performance, and field of application of the Model T-45 prime mover.

DESIGN REQUIREMENTS

The contract under which the Solar Model T-45 unit was produced had as a basic requirement that the prime mover be designed to drive a centrifugal water pump of the same rating as the pump used on the Navy's original Type P-500 portable water pump set. This rating, with sea water, is:

Flow	500 gpm.
Discharge pressure	100 psig.
Suction lift	16 ft.

With a power plant operating at the normal high speed of a gas turbine, it would appear only logical to design a high speed pump. However, it was found that this suction and flow rate cannot be achieved without cavitation at a pump speed much above 4500 rpm., which is the speed of the original pump. Therefore the original pump with some minor modifications was used, and the power plant was designed to suit.

Although standard ambient temperature for Navy gas turbines is 80° F., it was agreed that design calculations would be based on an ambient temperature of 120° F., to insure rated power under all operating conditions. At the rated flow conditions above, the pump requires an input of 47 hp. on the driven shaft. Hand starting was required, and Diesel fuel (Navy Specification No. 7-0-2e) was specified. A waterproof ignition system of maximum simplicity that would operate reliably without battery or external power was called for.

BASIC ARRANGEMENT

A single-stage, centrifugal compressor was chosen on account of the limitations on compressor pressure ratio imposed by the hand starting requirement. The influence of size effects, including those of Reynolds Number and leakage, and the necessity for holding weight to a minimum, also influenced this choice. Similar factors, as well as the need for simplicity and low manufacturing costs, dictated the use of a radial inflow turbine.

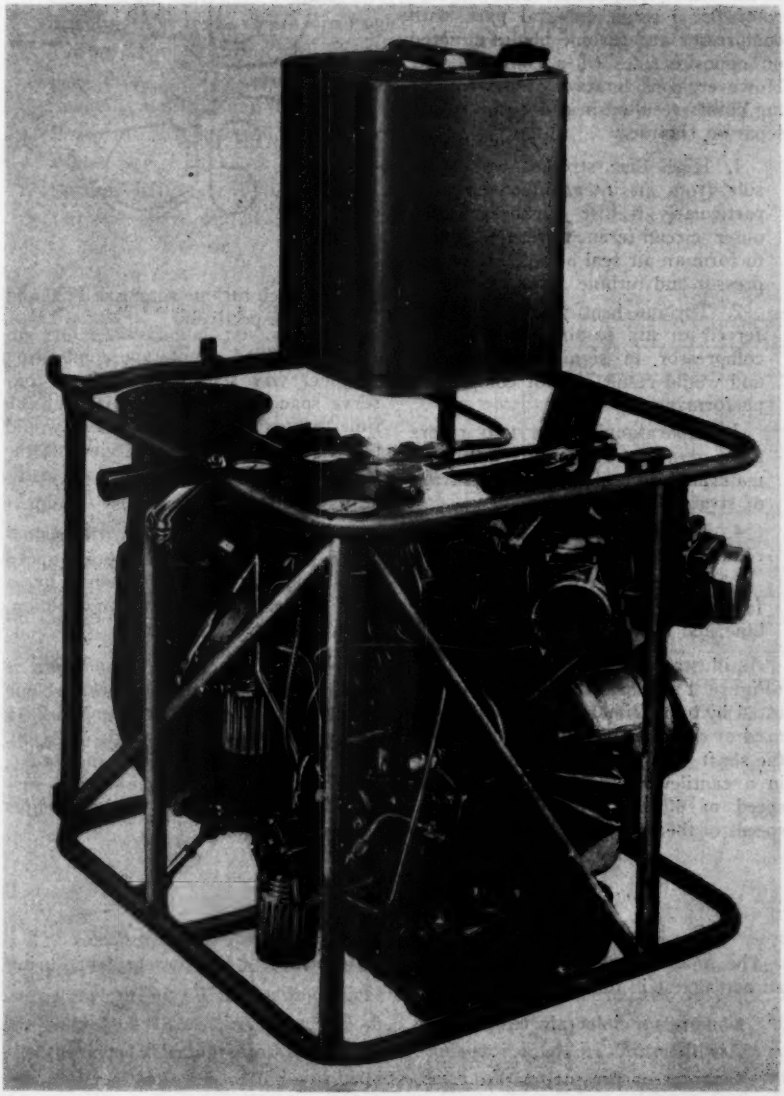


Figure 2.

At first it appeared desirable to construct the compressor and turbine rotors as a single integral part, with compressor and turbine blades situated on opposite sides of a common disc. However, consideration of the following points resulted in a decision to use separate rotors:

1. High disc stresses would result from an integral construction, particularly if disc thickness at the outer circumference were sufficient to form an air seal between the compressor and turbine areas.

2. Turbine heat would be transferred to air passing through the compressor in significant amounts, and would result in poor compressor performance.

3. Separation of the two rotors gives greater freedom in selection of materials, and a consequent saving of strategic alloys.

4. Machining time is considerably reduced by separation of the rotors.

5. Separating the rotors greatly facilitates development of the turbine and compressor components.

As illustrated in the section diagram (Figure 1), the two rotors are separated by approximately 0.5 inch, measured at the hub. They are mounted on the shaft adjacent to the roller bearing in a cantilever arrangement. At rated speed of 40,300 rpm., the peripheral speeds of the 6.97-inch compressor rotor and 7.42-inch turbine rotor are 1220 and 1300 fps. respectively.

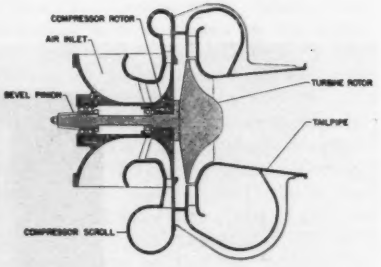


FIG. 1

The modified elbow-type combustion chamber was selected chiefly to conserve space. It was recognized at the time that extensive development would be needed on this part in view of the sparse design and operating data available.

For maximum simplicity, the reduction of turbine speed to the pump speed requirement was accomplished by a single spiral bevel gear and pinion, providing right-angle drive.

The arrangement of components is illustrated in Figure 2. The envelope dimensions of the set are: length, 26⅞ in.; height, 23½ in.; width, 23⅞ in. The dry weight, less fuel and tanks, is 165 lbs.; fuel for one hour's sustained operation, with tanks, weighs 112 lbs. additional.

PERFORMANCE

The following chart reflects operating performance of the power plant during an actual test in the shop:

Compressor inlet air temperature	81 F.
Air flow	2.35 pps.
Compressor pressure ratio	2.44
Compressor efficiency	74.5%
Combustion efficiency	95.0%
Turbine efficiency	78.0%

Turbine speed	39,800 rpm.
Compressor discharge temperature	316 F.
Turbine inlet temperature (average)	1142 F.
Tailpipe temperature (without cooling)	900 F.
Fuel flow	104 pphr.
Specific fuel consumption	2.22 p./hp.-hr.
Pump discharge pressure	103 psi.
Water flow	505 gpm.
Suction lift	18.7 ft.
Turbine shaft output	46.8 hp.

Other test runs were conducted to determine maximum power absorbed by the water pump with the discharge lines wide open and with minimum suction at the pump inlet. Test results were 770 gpm. at 82 psi., which is equivalent to 57 shp. Turbine inlet temperature increased to 1275° F. with a 6% decrease in fuel consumption, giving an sfc. of 2.1 p./hp.-hr. Turbine speed was held approximately constant during all test runs.

Figure 3 shows the fuel flow and shaft horsepower relationship in 80° F. ambient temperature. Increasing ambient up to 120° F. causes sfc. to increase by approximately 4% maximum.

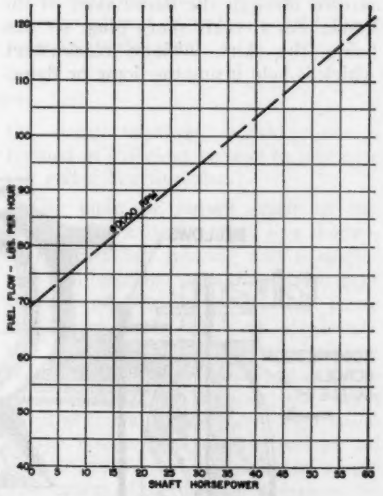


FIGURE 3.

COMPRESSOR

The centrifugal type compressor rotor is machined from a 25ST aluminum alloy forging. Blade leading edges are shaped to serve as an inducer. Both the turbine and compressor rotors are mounted back-to-back, with a seal plate between, on a common shaft which extends through the center of the compressor scroll and air inlet casing. A nine-blade, stainless steel compressor diffuser is bolted to the scroll. Compressor air is discharged through a bellows-type expansion joint to the combustor inlet. A butterfly valve in the scroll outlet provides a means of creating a low velocity area in the combustor during starting to facilitate ignition.

COMBUSTOR

Between the compressor scroll outlet and the turbine scroll inlet is a modified elbow-type combustion chamber, formed from Type 321 stainless steel in order to conserve critical materials. As shown in Figure 4, it differs from the normal gas turbine combustor in that there is no liner, but only a short, flared, dome-shaped flameholder set in the turbine scroll opening. An 8.3 gph., 80-degree spray angle Monarch nozzle admits fuel to the airstream through the flared skirt of the dome. An aircraft spark plug, set just below the skirt, initiates combustion, which is held inside the dome or flameholder.

Much cut-and-try development was required to obtain efficient combustion with Diesel fuel under the space and weight limitations. Six different configurations were fabricated and tested. Almost all of the evils associated with combustor design, such as poor efficiency, carbon accumulation, smoke, combustion instability, and hot spots, were met and overcome before the final design was evolved. The resultant configuration gives a pressure loss and combustion efficiency which compare favorably with can-type combustion chambers now commonly in use.

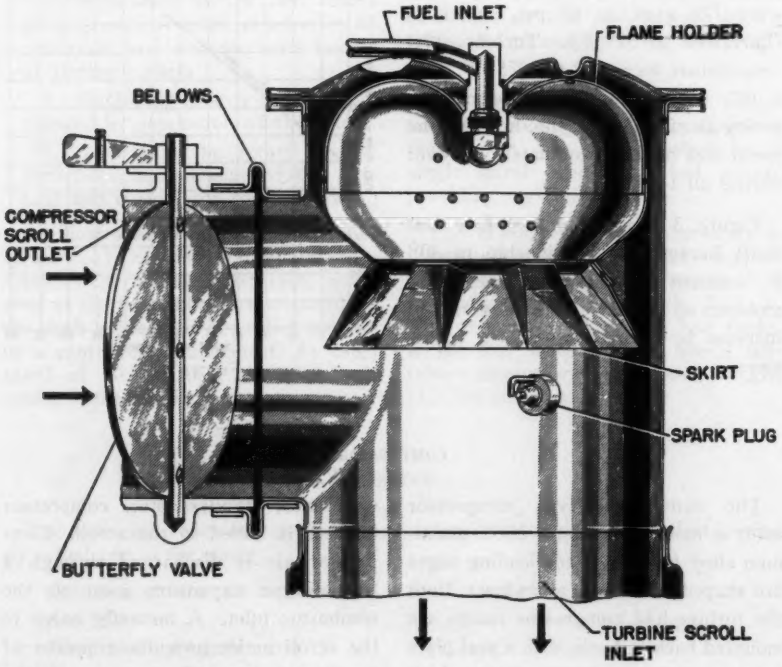


FIG. 4

TURBINE

As previously mentioned, the radial inflow turbine rotor is mounted on the shaft with the compressor rotor. Its blades are shaped to form an exducer in the same manner that the compressor blades are shaped to form an inducer. The rotor is encircled by the turbine scroll and nozzle, and a short tailpipe is centered behind the rotor hub. No exhaust diffuser is used, but a means is provided for cooling the exhaust gases, which will be described later. Hot gases pass from the combustor into the turbine scroll and are directed angularly inward by the turbine nozzle blade contour. Thence they whirl farther inward through the rotor and turn to exhaust axially through the tailpipe.

A Hastelloy B forging was chosen for the turbine rotor. This alloy contains no cobalt or chromium, it minimizes thermal distortion by its low coefficient of expansion in comparison with other high-temperature alloys, and is an excellent casting material. The rotor on the prototype model was machined, but future rotors will be investment cast in order to hold down manufacturing costs.

BEARINGS

Anti-friction ball bearings are used exclusively in the power plant, and since the pump impeller is mounted on the speed reducer output shaft, there are no bearings in the pump. Two main bearings support the turbine shaft in the air inlet housing. A matched pair of angular-contact ball bearings are located on the power output end of the shaft to take approximately 92 pounds of pinion gear thrust, and a single roller bearing is installed on the rotor end. Radial grooves cut in the mating surface of the outer races provide for pressure lubrication of the ball bearings. Oil from the pump passes through these grooves into the inner races. At the high operating speed of the main bearings, air-oil mist is formed in sufficient volume to lubricate the roller bearing also.

The main or output shaft of the speed reducer is supported in a double-row thrust ball bearing and a single-row ball bearing. Accessory drive gear shafts and the gear-type oil pump shafts are each borne in single-row ball bearings. Lubricating oil is directed through a jet nozzle against the out-of-mesh point between the input bevel gear and pinion. The high speed of the pinion here also causes air-oil mist to form and to maintain ample lubrication of the other bearings in the gear housing.

REDUCTION GEAR AND ACCESSORY DRIVE

Speed reduction between the 40,300 rpm. power unit and the 4428 rpm. water pump is accomplished by a 9.1 to 1 spiral bevel gear and pinion combination as shown in Figure 6. The pinion is secured to the turbine shaft where it extends into the gear housing, and meshes with the driven gear mounted on the speed reducer output shaft. The turbine axis is thus at a right angle to the output shaft. The pump impeller is mounted on one end of the latter, in the pump housing. The starter sprocket and clutch are mounted on the other end of it. Spur gears and a pinion on the output shaft drive the fuel pump and governor, oil pump, and overspeed shut-off valve. Gears, shafts, and bearings are enclosed in the gear housing, which also forms the lubricating oil reservoir and sump. The housing is supported between the drive end of the turbine and the water pump housing, with the oil pump and filter attached to it.

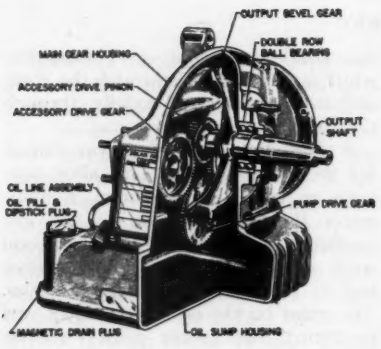
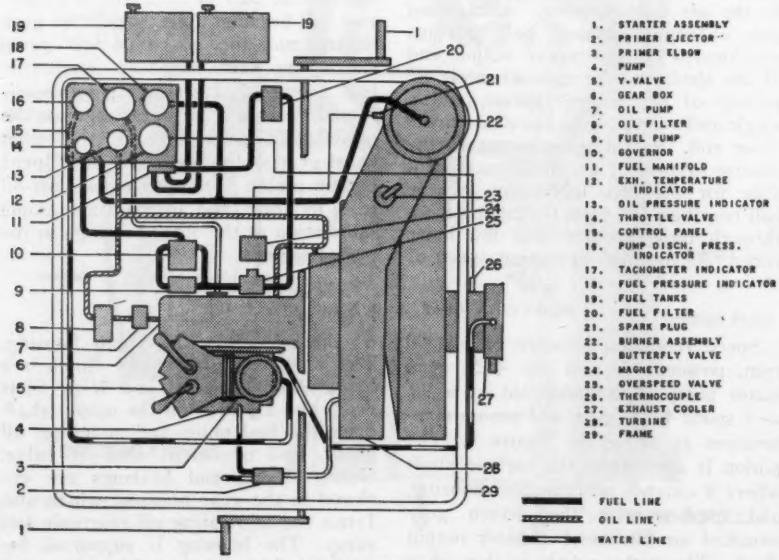


FIG. 8 CUTAWAY VIEW OF MAIN REDUCTION GEAR AND ACCESSORY DRIVE

FUEL SYSTEM

Fuel for one hour's sustained operation at full load is contained in two 7-gallon tanks. During operation the tanks are mounted on the tubular frame of the unit above the turbine tailpipe. Each tank is connected to a fuel manifold on the control panel by a short hose and self-sealing coupling. Fuel system arrangement is illustrated in Figures 2 and 5. When the turbine is in operation, fuel flows from the manifold through the fuel filter, overspeed shut-off valve, fuel pump and governor to the throttle valve and then to the fuel nozzle. At starting, only the right-hand tank can be used, as the left-hand tank covers the butterfly valve which is essential to starting procedure. After the start, however, either or both tanks can be connected. The self-sealing couplings allow either or both tanks to be replenished during operation without interruption of fuel flow.



SCHEMATIC DIAGRAM

FIG. 5

LUBRICATION SYSTEM

A gear-type oil pump with integral pressure relief valve draws lubricating oil from the reservoir, pressurizes it to a constant 20 psi., and delivers it at 0.75 gpm. to the turbine and reduction gear bearings. Scavenge oil passes by gravity back to the reservoir. A magnetic clean-out plug in the reservoir accumulates metal particles, and a micronic filter is installed on the pump outlet line.

IGNITION SYSTEM

Ignition is required only momentarily during the starting cycle. A magneto and spark plug are used to ignite the fuel-air mixture when the turbine has accelerated to approximately 4500 rpm. The magneto is equipped with a sprocket drive, connected by chain to a sprocket on the handcrank shaft. At 4500 rpm. turbine speed, the magneto is rotating at a speed capable of delivering sufficient ignition current to the spark plug. After ignition, cranking is continued until a speed of 11,000 to 12,000 rpm. is reached, at which time the engine becomes self-sustaining and accelerates further under its own power. The handcrank chain drives through an overrunning clutch at the reduction gear box, which precludes the turbine from driving the handcrank chain and shaft.

ACCESSORIES AND CONTROLS

A simplified control system governs sustained operation once the T-45 starting cycle has been completed. Exclusive of the fuel and oil supply systems previously covered, only three accessories and five instruments are used. The control system maintains rated turbine and pump output under varying ambient temperature conditions and during load changes resulting from the opening and closing of additional hose nozzles. It also protects the equipment from overspeeding due to loss of prime, cavitation, or malfunction. This is automatically accomplished without attention from the operator. In addition, the system indicates fuel, lube oil, and water pump discharge pressures at all times. The location and inter-connection of control components is brought out in Figures 2 and 5 and the function of control components is given below.

The fuel pump is a gear-type unit driven off the reduction gear. Its normal rating is 3250 rpm., at which it delivers 55 gph. of fuel to the governor inlet (approximately three times the maximum requirement of the power plant). A booster pump is not needed at starting or otherwise.

The governor, mounted on and driven by the fuel pump, is a droop or proportional control type. After the start it regulates fuel flow to maintain turbine speed at 40,300 rpm., and bypasses excess fuel back to the pump inlet. A minimum flow valve always assures enough fuel to prevent loss of combustion, even with the governor bypass wide open at extreme overspeed.

The overspeed fuel shut-off valve, driven off the reduction gear, is a spring-loaded, flyweight-actuated shut-off in the fuel line between the tank and the pump. It is opened manually prior to the start and held open by a latch. If the turbine overspeeds, the flyweight releases the latch and the spring moves the valve closed.

The exhaust temperature indicator, mounted on the control panel, is wired to a chromel-alumel thermocouple in the tailpipe and registers exhaust gas temperatures at all times.

The fuel, lube oil, and water pressure gages indicate outlet pressures of the three pumps.

The tachometer indicator is connected to the reduction gear output shaft and indicates pump impeller speed.

STARTING

So far as is known, the T-45 is the first gas turbine ever built which is started by hand. Careful thought and effort were expended on design features which would minimize the exertion necessary to bring the turbine up to speed by hand cranking. The feasibility of the method was in doubt until the mechanism was actually constructed and tried. It proved to be practical, however, and the main contributing factors are as follows:

First, a moderate pressure ratio was selected, because normally the lower the pressure ratio of a gas turbine engine, the lower the speed at which it becomes self-sustaining and the less effort is required to crank it. Another factor in choosing a pressure ratio lower than those possible with this type wheel and reasonable efficiencies, was a design compromise forced by limiting turbine tip speed to an acceptable value for the material strength at the turbine operating temperature.

Second, mechanical losses were avoided by eliminating bearings where possible, and using anti-friction type where bearings were essential.

It was assumed during initial design that hand starting would be significantly aided by the use of a manual butterfly valve in the compressor discharge. This valve would be closed, taking pumping load off the compressor, during acceleration up to ignition speed, and then opened for the balance of the cycle. In practice, however, this effect was found to be of little help, although the valve was retained to insure positive ignition at all times.

The chain and sprocket arrangement between handcranks and pump shaft, with the gear-and-pinion pump-to-turbine connection, provides an 81 to 1 speed increase in the starting train, and leads to a practicable torque on the cranks during accelerations. When turbine speed exceeds crank input speed, a Formsprag overrunning clutch on the pump shaft disengages the starting device.

To clarify this unconventional starting method, a simplified step-by-step sequence of starting operations is added below.

1. With the butterfly valve closed, cranking is started and continued until pump speed reaches 500 rpm. Then the throttle is opened.

2. At this speed the magneto is delivering ignition current to the spark plug, and combustion commences. The butterfly valve is opened immediately.

3. Cranking is continued, with throttle adjustments made manually to keep turbine exhaust temperature within the specified starting range, until 1200 rpm. pump speed is reached.

4. From this point on the engine is self-sustaining, and with continued throttle adjustments per (3) it will accelerate up to governed speed without cranking.

EXHAUST COOLING

Because exhaust gases from the turbine cannot always be directed away from working areas and surrounding structure when operating the fire pump, the turbine tailpipe is equipped with a cooling device. This consists of a hollow cone type, non-clogging spray nozzle in the tailpipe which introduces water from the pump discharge into the exhaust stream at about 3 gpm. Exhaust temperature is held down to 350-400° F. by this method.

EJECTOR AND HAND PRIMING

Evacuation of air from the pump and suction line is accomplished by an ejector type primer on the pump housing, shown in Figure 7. When the pump is at normal speed, opening the ejector allows compressor discharge air to flow through the ejector valve and aspiration chamber to atmosphere; this draws air from the suction line and creates a vacuum sufficient to raise water up to the pump impeller. With a 21 psig. discharge air pressure (normal speed), a suction of 23.25 in. Hg. is created. During factory tests the pump was put in operation at full capacity by this primer in 45 sec., using a 4-in., 30-ft. long suction line with a 16-ft. lift. As an alternative, hand priming can be accomplished with the pump at rest by pouring water into the housing with a hose or bucket until the suction line is filled. A valve in the pump outlet provides an orifice for this type of priming.

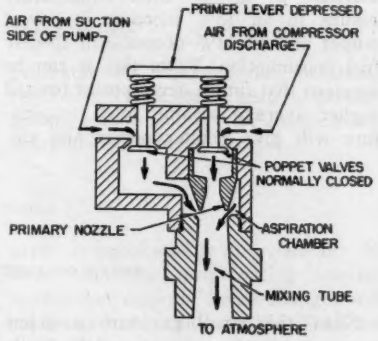


FIG. 7

OPERATING EXPERIENCE

Final shop testing of the T-45 was begun in January 1950. Initial no-load starts were made with battery ignition, using gasoline for fuel. After a proper sequence of starting operations was established, magneto ignition and Diesel fuel were used. The only difference in operation noted from this change was a slightly slower acceleration below 20,000 rpm. Above 20,000 rpm. there was no appreciable difference.

After several no load speed runs were completed to check temperatures and proper functioning of controls and accessories, full load tests were begun. The data obtained on the first full load run was substantially as given in the foregoing discussion of performance. Through the following weeks all components were checked and tested. The power plant was disassembled 15 times for inspection, repairs, and minor changes. In all, 278 starts were logged with a total operating time of 38.3 hours, 11.8 hours of which were at loads ranging from 7 to 125% of rated load. The following data were taken on one of the final runs.

T-45 POWER PLANT

Turbine rpm.	*Amb. Temp. deg. F.*	*Turb. Inlet Temp. (avg.)*	*Turb. Exh. Temp. deg. F.*	*Pump Flow, gpm.*	*Pump Press, psi.*	*Suction in. Hg.*	*HP.*	*SFC php.-hr.*
40,000	73	1192	950	527	109	—1.0	48.3	2.38
39,400	77	1217	1000	750	80	—19.5	50.0	2.32
40,000	75	1275	1045	769	82	—20.0	57.2	2.10

These data show that an 83 deg. F. increase in turbine inlet temperature results in an 18% increase in power output and an 11% decrease in specific fuel consumption. From this it can be foreseen that future development toward higher average turbine inlet temperature will give greater power and efficiency without change in overall size and weight. The present design limit of 1300° F. maximum turbine inlet temperature, based on the use of non-critical materials, could be raised substantially by employing critical high-temperature alloys, or by turbine wheel cooling.

TREND OF FUTURE DEVELOPMENT

Since this small gas turbine prime mover has performed successfully in its first application, it appears logical to expand its potentialities with respect to improved performance. A feature to be improved in virtually every small gas turbine is the relatively high specific fuel consumption which becomes especially apparent during long continuous operating periods by canceling the weight advantage of the basic engine. Several methods present themselves, the most obvious being improvement in turbine and compressor efficiencies. This improvement is of an evolutionary nature and continues as long as there exists a demand for the engine. The more difficult methods of achieving better performance are those which are common with larger gas turbine plants and which have received discussion in technical papers throughout the world. Beyond the raising of operating temperature and pressure, these remedies consist of the addition of external heat exchangers and supplementary burners such as are used for reheat.

The field of turbine cooling mentioned above under Operating Experience appears to be especially attractive for marine units since liquid cooling does not impose the critical weight problem normally associated with aircraft installations. In the cooling of gas turbines two philosophies can be followed; the first, that which eliminates or at least minimizes the use of high temperature alloying agents with no sacrifice in performance, and, second, that which permits gas turbine operation at the stoichiometric ratio with the attendant high temperatures utilizing present materials. Ideally speaking, the goal is to operate gas turbine marine plants at extremely high temperatures using nothing but common, uncritical materials. While it is recognized that this goal is unattainable in the immediate future, every step toward its achievement represents an improvement which can be translated directly into terms of improved gas turbine performance.

LIFE

The question of total service life and operational periods between minor overhauls is one which is highly important in any new engine development. At the present time, there is insufficient data from which definite life characteristics can be determined. However, the design life goal of at least 500 hours total operating time with 50 hours minimum time between replacement of the combustion chamber liner or bearings appears to be reasonable and should be achieved without difficulty. This statement is, in the opinion of the authors, a safe one in view of the relatively low operating temperatures of the turbine rotor.

OTHER POSSIBLE APPLICATIONS

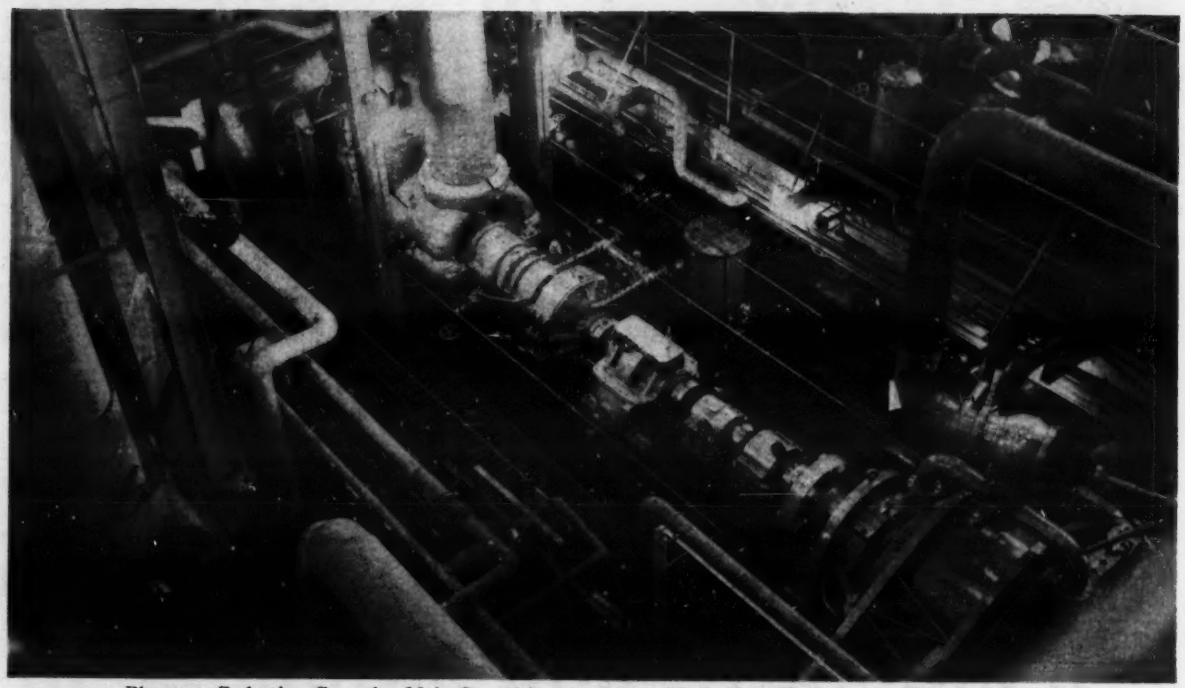
Typical applications in addition to that described above include the following: Automotive-mounted fire-fighting equipment, light-weight portable generator sets, superchargers for Diesel engines, portable space heaters, salvage pumps, compressor drives, engine starters, and fuel transfer pumps.

CONCLUSIONS

In addition to the aforementioned applications, numerous industrial uses for the T-45 are apparent. The present specific fuel rate points toward service of an emergency nature where each run is of relatively short duration. For example, immediate use as generator drive in emergency circuits of hospitals and merchant ships plus emergency compressor drive in small freezing plants is suggested.

As usage widens, the unit cost will become comparable to other prime movers of the same horsepower. It is expected that maintenance and spare parts requirements will eventually be lower than for reciprocating internal combustion engines. Some changes will continue to be made to the T-45 but these consist largely of refinements in the interests of production facilitation and lowered cost rather than basic design revisions.

The advantages mentioned above, coupled with the widely publicized inherent gas turbine characteristic of multiple fuel adaptability make the T-45 gas turbine engine a promising entry in the field of successful prime movers.



Planetary Reduction Gear, for Main Propulsion, on test at U. S. Naval Boiler and Turbine Laboratory

RUBBER HOSE USED BY THE NAVY

E. A. BUKZIN

THE AUTHOR

recevied his Bachelor's degree in Chemical Engineering from the College of Engineering of New York University in 1939. He was employed as an illumination engineer in New York during 1940 before coming to Washington, D. C. After obtaining sufficient graduate credits in Naval Architecture, he came to work for the Bureau of Ships in May 1941 in that capacity. In August 1943, he transferred to the Rubber Branch of the Research and Development Division where he is presently employed as a Senior Materials Engineer. He was elected Treasurer of the Washington Rubber Group, an affiliate of the American Chemical Society, for the year commencing September 1949.

INTRODUCTION

The use of rubber and rubber-like materials on Naval vessels is seldom recognized in the true light of its importance since it is but a small part of the ship with respect to cost and weight. As indicated by Werkenthin, Bukzin, Greenleaf, and Levene, "Some Uses of Rubber in Naval Vessels," JOURNAL OF AMERICAN SOCIETY NAVAL ENGINEERS, Volume 62, No. 4, November 1950, its use is varied and valuable. Rubber hose used in our Navy is a general title for many applications of basically different equipment as will be discussed herein. There are only about eight basic types of construction of rubber hose in production in the rubber industry today, which, with slight modifications, are used in various applications. Variations within these constructions are almost too numerous to mention and are used as the intended service requires. Just about all of these types are used by the Navy in its air, ship, and shore operations in one form or another. It must be pointed out that with very few exceptions, the various hoses used were developed by the industry. This follows, since hose production involves highly specialized, complicated and expensive equipment as well as highly skilled artisans and designers.

HOSE TYPES

Among the many types of hose used by the Navy are several peculiar to it. They include hose for submarine rescue chamber service, for diving operations, for spraying hot plastic paint to ship bottoms, for salvaging ships and pumping out flooded compartments, for fueling at sea, etc.

The various types may be generally differentiated by the difference in construction and manufacture as follows:

(a) *Mandrel built, plied and wrapped.* The inner tube is applied over a mandrel either by lapping a sheet of material or by applying an extruded tube over the mandrel. The carcass or strength member of the hose is developed by applying rubber impregnated cloth (usually cotton duck or rayon) cut on the bias in sufficient plies to develop the burst strength desired. The rubber cover is then applied over in a manner similar to the tube. The whole assembly is then wrapped under tension with a cotton tape which has been wetted with water. The mandrel and hose assembly is then placed in a heater and vulcanized at a temperature ranging from 275°F to about 350°F. The tape dries out and shrinks thus developing a pressure on the hose and knitting the mass of tube, impregnated cloth, and cover together. After cure is complete, the tape is removed and the hose is literally blown off the mandrel with compressed air. The cotton tape fabric impression may be observed for hose of this type.

(b) *Mandrel built, braided, and wrapped.* The technique is the same as (a) above except that the carcass consists of one or more braids of cotton, asbestos, wire or rayon fibres or combinations of these materials. The advantage of this hose lies in lighter weight and speed of production. It involves the use of very costly and complicated braiders.

(c) *Lead press cured.* Hose made for lead press cure is usually made for large volume consumption. The tube is extruded from a reasonably "stiff" compound so that the carcass can be braided or woven around it without collapsing it. The cover in turn is extruded over the carcass. This is a continuous process which carries the hose to the lead press where lead is fitted as a jacket around the hose cover from a molten state. As it leaves the lead press the hose is reeled up to the desired length and cut. The hose is filled with water through suitable fittings and placed in a steam autoclave. The water is kept under pressure and thus expands the whole assembly against the lead cover while the heat effects a cure. Hose cover designs are applied to hose made in this fashion as may be observed on most garden hose used domestically.

(d) *Wire reinforced smooth bore mandrel built hose* is usually manufactured in large diameters for suction or suction and discharge service. Its manufacture is essentially the same as (a) above except that one or more wire helices are built into the carcass plies to add the required strength for suction working and/or discharge pressures. This hose is essentially hand built and quite expensive. Nipples to which couplings or flanges may be threaded are generally built into the ends of such hose. The wire or wires serve for static grounding and are bound and brazed onto the nipples in an intricate manner for strength at this point.

(e) *Wire reinforced rough bore mandrel built hose* is manufactured essentially in the same manner as (d) above and for essentially the same service. It differs in that, in lieu of the initial application of rubber tube over the mandrel, a wire helix is installed and a rubber treated cloth breaker is applied over it before subsequent plies of rubber,

fabric, wire, etc., are applied. Its cost is somewhat less than (d) and can be made somewhat faster. Pressure drops are greater, however, due to additional turbulence created by the inner wire helix.

The built-in nipples are generally used with hose of this kind as well as for 4" I.D. sizes and larger.

(f) *Wire braid mandrel built hose* consists of a tube either lapped or extruded over a mandrel of the desired inside diameter of the hose. One or more layers of wire is braided over the tube with rubber layers separating the braids. A rubber cover is applied over all and a "wrapped" cure in a heater follows. Such hose is usually limited in inside diameter to less than one inch. It is designed for very high internal working pressures as are encountered in hydraulic lines, grease lines, CO_2 fire extinguishers, etc.

(g) *Loomed (fire) hose* is generally made by looming one to three cotton jackets to be pulled over a rubber lapped or extruded tube already installed on a suitable mandrel. The jacket(s) are wetted down and serve in lieu of wrapping when placed in the heater to effect a cure of the rubber tube.

(h) *Mandrel built spiral wound and wrapped hose* is of relatively recent design and involves spiralling up to four layers of cord over a tube applied to a mandrel. This hose has been spliced into 2000-foot lengths from standard fifty-foot lengths. It is lightweight hose which can be made in large diameters at high speed but is limited to discharge service only.

The above represent the major hose types, variations and combinations of which are not uncommon. Except as indicated for the heavy large diameter hoses, fittings are applied after the hose is cured. The type of fitting used depends on the expected service as well as the hose construction and competition. The subject of fittings involves much more than this paper will attempt to cover.

APPLICATIONS

Submarine rescue chamber hose was designed for air supply to and venting from the rescue chamber. The supply hose has a one-inch inside diameter and utilizes a bias wrapped construction with no wire reinforcement. The vent hose is one and one-quarter inches in inside diameter and utilizes a wire helix reinforcement to resist collapse. The vent hose tends to collapse since it is under atmospheric pressure internally and may be at a depth of some 1200 feet. Navy Department Specification 33H5 requires that this hose show no more than a 30% reduction in inside diameter when subjected to a 600-pound-per-square-inch hydrostatic crushing pressure. The retention of air in the supply hose at this same pressure offers a difficult problem of overcoming leakage from the standpoint of an industrial hose length inspection test which will be discussed later in this paper.

Divers Air Hose. Divers hose, in accordance with Navy Department Specification 33H9 in one-half inch inside diameter and of a bias wrapped construction is fraught with similar leakage problems, but again we are dealing with high air pressures. The problem is far more acute here, however, since divers will object to even the sight of air bubbles that may cling to the hose cover. Yet, their feelings on such matters cannot be disputed particularly since burst failures have occurred.

Hot Plastic Paint Hose. This hose is essentially peculiar to Naval use in that it conveys the hot plastic anti-fouling

paint from the heating vessels to the spray nozzle for application to ship bottoms. It is procured in accordance with Navy Department Specification 33H26 as it is necessary to keep the paint at about 250°F. This is done by special fittings attached to each end of the hose length to which is attached a heating wire element strung through the bore of the hose. The electrical circuit is completed by a wire taped to the outer cover of the hose. Incidentally, this is the one hose we specify to be made with a glass fibre reinforcement.

Salvage Hose. Salvage hose in accordance with Navy Department Specification 33H30 was developed for use in the salvage of many ships that were put out of commission during the Pearl Harbor incident. This hose is very similar to rough bore oil suction and discharge hose, but is designed for additional flexibility required to snake it through hatches and man holes to flooded compartments.

Fueling Hose. Light weight, large diameter oil and gasoline suction and discharge hose for fueling at sea included both rough and smooth bore types. As a result of several full-scale pumping tests, the detailed data of which are published, Werkenthin, James and Morris, "Comparison of the Carrying Capacities of Rough and Smooth Bore Oil Suction and Discharge Hose Under Simulative Service Conditions," "Rubber Age," September 1947, and Kaufman, R. E., "Flow Tests Through Fueling Oil Hose," "The Petroleum Engineer," November 1949, rough bore fueling hose in accordance with Navy Department Specification 33H4 has been largely discarded due to the improved efficiency of the smooth bore type. The latter may be procured in accordance with Navy Department Specification 33H17 for extra heavy duty and 33H21 for normal service. Should the emergency become acute, procurement of rough bore hose may be resumed due to the greater ease of manufacture and the smaller quantity of rubber used in its construction.

Long Length Fueling Hose. Such hose involves a new experimental technique of transferring fuel from ship to shore utilizing continuous lengths of 4" discharge hose as much as two thousand feet long. This 4-ply spiral cored hose is made with vulcanized flexible splices between each conventional 50-foot length. It may be of interest to know that recognition has been made of contamination to fuel by hose tube material. As a result of a detailed study of compounding ingredients, the contaminants were determined to be the various phosphate plasticizers. These phosphates react with the tetraethyl lead additives, thus reducing the octane rating of the gasoline. A method for identifying the presence of phosphate plasticizers has been incorporated into all our fuel hose specifications to assure that these plasticizers are not used.

Steam Hose. A major service deficiency in most commercial steam hose has been noted in hose advertised for use with 200-pound-per-square-inch saturated steam. Such hose used in our shipyards serve as a flexible connection from the shore steam plant to ships' piping systems when the latter are tied up at docks. It is used for heating these vessels in the winter and for cooking and cleaning, as well as to warm up engines and run auxiliary machinery. The hose in question is usually subjected to rough treatment. Such treatment may involve water hammers formed by condensation in a loop of the hose as it dips into the water between the dock and ship when the former is flooded. It is required to convey steam continuously at a pressure as high as 170 p.s.i. from 90 to 100 days. Failure has occurred in as short a time as four hours and with the exception of two manufacturers' products which utilize braided wire and asbestos

constructions, failures generally occur in seven to thirty days.

Paint and Dope Spray Hose. This hose is used in large quantities by the Navy and is specifically designed for dope and paint spray application. It utilizes a Thiokol tube to resist lacquer and paint thinners and probably is the only hose manufactured with this type rubber today. Navy Department Specification 33H19 under which this was procured has been superseded by Federal Specification ZZ-H-521.

Fire Hose. The Navy uses conventional double-jacketed fire hose in enormous quantities, particularly the 2½-inch size in accordance with Federal Specification ZZ-H-451. Its service life does not usually compare well with that obtained by civilian fire departments. It has been determined that civilian fire departments carefully clean and dry their hose after each operation. Likewise, they maintain the hose under careful storage conditions which are not always feasible for us. Accordingly, difficulty has been experienced with mildewing of the cotton jackets of fire hose used by the Navy. As a result, our fire hose rarely lasts more than five years, and sometimes lasts as little as six months in some of the hot humid areas of the world, as compared with a life of 10 to 20 years for civilian fire departments.

Sand Blast Hose. Sand blast hose is used in large quantities at our shore establishments. It is procured under Navy Department Specification 33H24, and it is not too unlike that in commercial service. The difference is in its material requirements that specify synthetic rubber in lieu of pure gum natural rubber tubes.

Other Hoses. Other hoses used by the Navy are of a more conventional type. They include fire extinguisher hose which utilizes a wire braid construction; pneumatic and welding hoses which utilize both wrapped and braided constructions; and gasoline discharge and suction-and-discharge hose purchased for all three services under the same specifications for the past several years. The latter is largely manufactured using the Chernak loom process. Self-sealing fuel line hose and high pressure hydraulic hoses used in Naval aircraft are the same as those used by the Air Force.

General Developments. After several years of detailed inspection of ships putting-in to various shipyards by Navy laboratories' personnel, we have recognized the need for a requirement for a high grade neoprene cover for all rubber covered hose. Neoprene is particularly resistant to weathering and abrasion when properly compounded. This need stems from the breakdown of hose covers by abrasive wear as well as ozone and sunlight cracking and crazing. Admittedly, the handling and stowing techniques used by operating personnel do not always lend themselves to increasing the longevity of the life of rubber hose. Instructions for overcoming this problem have been of little or no avail. In order to repair cracked and chipped hose covers, advantage was taken of a liquid polymer polysulfide rubber compound early in the development of these materials by the Thiokol Corporation. A paste-like, self-curing compound was designed for making repairs to such rubber hose covers. However, little success was obtained in getting the forces afloat and ashore to use this material despite the simplicity of its application.

With respect to the air leakage problem on divers' hose and submarine rescue chamber supply hose mentioned above, a means of inspecting each length of hose at the factory to insure satisfactory service is required. Hydrostatic tests are not adequate, particularly where small hard foreign matter inclusions may be present in the tube.

Such particles may be in the form of contaminants found in smoked sheet bales where hevea rubber is used or those picked up in the milling and calendering operations in the rubber factory. Likewise, thin spots caused by stretching on mandrels may reduce the effectiveness of the tube to retain the air. The Bureau of Ships is engaged in trying to develop a test that might be successfully used in the factory. This may consist of an air test with suitable safety precautions to eliminate possible injury to testing personnel and will also limit delays in delivery. The possibility of a high voltage electrical test for finding flaws in the hose tube is likewise under serious consideration. In all probability, the specifications resulting from this work will require a foolproof test and probably will specify an increased tube thickness and preclude the use of Hevea (natural rubber) stocks.

Summary. In summary, this paper attempted to briefly describe some uses the Navy has for the various types of rubber hose. These types encompass all the major hose construction and manufacturing techniques. Some of the immediate needs and programs have been mentioned to point up how applied research and development has and will continue to be of prime importance in this field.

THE WAVE DRAG OF SHIPS

MAX OTTO KRAMER

THE AUTHOR

was born on the 8th of September 1903 in Koeln, Germany, graduated in April 1926 as electronic engineer at the technical college, Muenchen, and received his doctor-diploma at the technical college, Aachen, in June 1931.

By the time of his doctor promotion, he switched from electronics to aerodynamics, was assistant at the aerodynamic institute of the technical college, Aachen, and joined after that the German Institute for Aeronautics, Berlin, where he finally headed the aerodynamic institute.

Being an expert in certain aerodynamic fields, the American Navy requested him for work in the United States after the war. Since then he has been working at the Naval Air Development Center, Johnsville, Pa., for almost five years.

INTRODUCTION

Any slender body traveling far below the water surface has practically zero form drag. The remaining drag is due to friction of the fluid on the body's hull and can easily be calculated with sufficient accuracy.

If the body is partly submerged, the dynamic pressure along the hull raises and lowers the water surface about the body. The energy consumed for the raising and lowering of the water surface is usually lost since it is converted into wave energy and emitted in all directions over the water surface. Since energy is lost, there must be a corresponding drag on the body. This drag, due to wave generation on an originally smooth water surface, is called wave drag.

As far as the author knows, the literature dealing with the wave drag of ships offers nothing but empirical formulas, while the theoretical solution of the problem is considered hopelessly complex. The following derives a simple formula from a theoretical consideration and checks its value by comparison with official data of ships of the United States Navy.

THE CALCULATION OF THE WAVE DRAG

It may first be assumed that the shape of the ship is a slender body of revolution with elliptical cross section (Figure 1). For this body in an entirely submerged state, the average velocity increase at its contour is known to be

$$\Delta\overline{\vartheta} = \frac{B}{2L} \times \vartheta_0 \qquad (1)$$

where

B = Maximum thickness of the body (beam)

L = Length of the body

ϑ_0 = Velocity of the body

The average dynamic pressure due to $(\Delta\overline{\vartheta})$ is

$$-\Delta\overline{P} = d \times \vartheta_0 \times \Delta\overline{\vartheta} = -\frac{B}{L} \times q \qquad (2)$$

where

d = Density of the fluid

q = Ram pressure = $d/2 \times \vartheta_0^2$

If the body is half submerged (Figure 1), the average dynamic pressure $(\Delta\overline{P})$ will not be altered as long as the height of the waves is small in comparison with the dimensions of the body. The height of the waves generated will be

$$\Delta\overline{H} = \frac{\Delta\overline{P}}{s} \qquad (3)$$

where

s = Specific weight of the fluid

The force due to the raising or lowering of the water surface exerted on a vertical plate is

$$F = L' \times \Delta\overline{H} \times \frac{\Delta\overline{P}}{2} \qquad (4)$$

where

L′ = Length of the plate in the horizontal direction

Since the raising and the lowering of the water surface cause the same energy loss due to wave generation, the positive or negative sign of $(\Delta\overline{P})$ can be neglected. Considering both sides of the body and its great slenderness assumed

$$L' = 2 \times L \qquad (5)$$

$$F = L \times \Delta\overline{P} \times \Delta\overline{H} \qquad (6)$$

As far as drag is concerned, only the component of the force in the direction of motion of the body is of importance. Thus the force calculated must be multiplied with the average angle of attack of both sides of the body

$$\overline{\alpha} = \frac{B}{L} \qquad (7)$$

The force thus determined is the first approximation of the wave drag of a body of revolution with elliptical cross section (Figure 1)

$$D_w = \frac{1}{s} \times \left(\frac{B^2}{L}\right) \times q^2 \times B \qquad (8)$$

The formula indicates exponents of (B) and (L) greater than unity. Thus the use of linear averages for $(\Delta\overline{P})$ and $(\overline{\alpha})$ must lead to considerable errors if (ΔP) and (α) along the contour of the body are not approximately constant for a given speed and shape.

Neither (ΔP) nor (α) is nearly constant for an elliptical cross section of the body (Figure 1). Thus the shape of the body must be modified to make (ΔP) and (α) approximately constant.

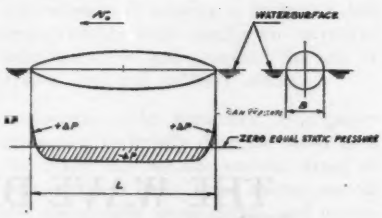


FIG. 1 THE BODY OF REVOLUTION WITH ELLIPTICAL CROSS SECTION HALF SUBMERGED IN WATER. ΔP = DYNAMIC PRESSURE ON THE CONTOUR.

The proper method to accomplish that is to add two cones to an elliptical middle section (Figure 2). This alteration lowers the pressure and angle of attack peaks near both points of the body. Simultaneously, the shape of the body comes closer to that of actual ships.

There is a certain length of the cones that makes $(+\Delta\overline{P})$ at the cones equal to $(-\Delta\overline{P})$ at the elliptic middle section. This length will make (ΔP) and (α) as constant as possible and permit the use of the wave drag formula derived. It may be assumed that the addition of the cones does not affect $(-\Delta\overline{P})$ at the elliptic middle section, an assumption which is tolerable for an approximate theory.

$(+\Delta\overline{P})$ at the cones can be derived from the theory of wings. The gradient of the lift coefficient of the wing with infinite span is

$$d(C_L)/d\alpha = 6 \qquad (9)$$

One third of this lift is due to the pressure side of the wing

$$d(C_L)/d\alpha\ (+\Delta P) = 2 \qquad (10)$$

The three dimensional equivalent of the pressure side of an infinite wing is the cone. The cone, therefore, generates half the specific pressure of the infinite wing

$$d(C_L)/d\alpha_{(cone)} = 1 \qquad (11)$$

The angle of attack of the outside of the cone is

$$\alpha = (B/L)_{\ cone} \qquad (12)$$

If (L) refers to a double cone corresponding to the both points of the ellipse. The average pressure on the outside of the cone is

$$+\Delta\overline{P}_{(cone)} = B/L\ (_{cone}) \times q \qquad (13)$$

since it shall be

$$(+\Delta\overline{P})_{cone} = (-\Delta\overline{P})_{\ ellipse} \qquad (14)$$

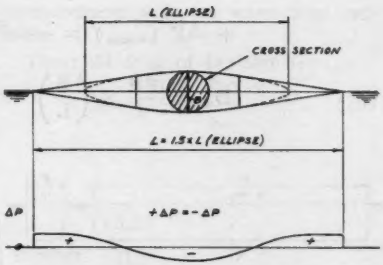


FIG. 2 BODY OF REVOLUTION CONSISTING OF TWO CONES AND AN ELLIPTICAL CENTER SECTION. ΔP = DYNAMIC PRESSURE ON THE CONTOUR.

$$(B/L)_{\ cone} = (B/L)_{\ ellipse} \qquad (15)$$

A simple geometrical calculation indicates that this requirement is met when

$$L_{\ (cones\ plus\ elliptical\ middle\ section)} = 1.5 \times L_{(ellipse)} \qquad (16)$$

This modification of the elliptical cross section is necessary to make the wave drag formula applicable. The 50 per cent increase in length must be considered in the formula. Thus the wave drag of a body of revolution consisting of two equal cones and an elliptic middle section (Figure 2) submerged to its center line is

$$D_w = \frac{2.25}{s} \times \left(\frac{B}{L}\right)^2 \times q^2 \times B \qquad (17)$$

Actual ship hulls seldom correspond with this idealized shape. They rather look like wedges with rounded off edges. Thus the wave drag of a hull, consisting of two wedges and a rounded off middle section (Figure 3), may be calculated too, in order to enclose the wave drag of actual hulls between theoretical limits. If the draft is assumed to be comparatively great, the upper limit of the wave drag is simply the two dimensional result of the formula found before for a body of revolution. In this case it is

$$+ \Delta\overline{P}\ (_{wedge}) = -\Delta\overline{P}\ (_{ellipse}) = 2 \times \frac{B}{L} \times q$$

$$D_w = \frac{9.0}{s} \times \left(\frac{B}{L}\right)^2 \times q^2 \times B \qquad (18)$$

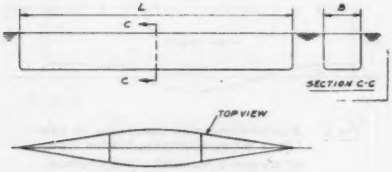


FIG 3 BODY WITH PARALLEL SIDES CONSISTING OF TWO WEDGES AND AN ELLIPTICAL CENTER SECTION. (DRAFT COMPARATIVELY GREAT SO THAT THE DYNAMIC PRESSURE IN THE WATER-LINE APPROACHES THAT OF THE TWO DIMENSIONAL FLOW)

Since actual ships have neither the true three dimensional shape (Figure 2) nor the true two dimensional shape (Figure 3), it may be concluded that the most probable coefficient of the wave drag formula will be the geometrical middle of both extremes. Thus the approximate wave drag formula for actual ships is expected to be

$$D_w = \frac{C}{s} \times \frac{B^2}{L} \times q^2 \times B;\ C \text{ approx.} = 4.5 \qquad (19)$$

where

D_w = Wave drag (lbs)

s = Specific weight of the fluid (lbs/cu. ft.)

B = Beam of the ship in the water line (ft)

L = Length of the ship in the water line (ft)

q = Ram pressure $= \frac{d}{2} \times \phi_0^2$ (lbs./sq. ft.)

d = Density of the fluid $\frac{(\text{lbs} \times \text{s}^2)}{(\quad \text{ft}^4 \quad)}$

ϕ_0 = Velocity of the ship (ft/s)

C = Constant ranging from 2.25 to 9.0 mainly depending on the degree of approach of the actual shape towards one of the two extremes considered.

COMPARISON OF THE FORMULA WITH OFFICIAL DATA

Each ship of the United States Navy is tested after completion. This test is made in deep and calm water. The result of this test shows the maximum speed of the ship and the power required to reach this speed. Navy catalogs show these data together with the displacement and the ratio (L/B) in the water line.

The appendix contains these data for seven battleships, four cruisers, and four destroyers. Only 15 ships were

chosen out of the great number of data because many ships are so similar that groups can be represented by a single ship. For each ship, the following calculation is necessary to compare the measurement with the wave drag formula.

The total drag of the ship is

$$D = \frac{P \times 550 \times e}{\vartheta_0} \qquad (20)$$

where

D = Total drag of the ship (lbs.)

P = Driving power (measured) (H.P.)

e = Propulsion efficiency

ϑ_0 = Velocity of the ship (measured) (ft/s)
(1 knot/H = 1.7 ft/s)

The propulsion efficiency is not measured and is therefore assumed to be constant

e = Const. = 0.7

The total drag is the sum of the friction drag (D_f) plus the wave drag (D_w)

$$D_w = D - D_f \qquad (21)$$

Since the calculation proves that the friction drag is only 10 to 20 per cent of the total drag, the simplest method can be used to determine the friction drag

$$D_f = C_f \times S \times q \qquad (22)$$

where

S = Surface of the ship hull (ft^2)

$$S = 0.8 \times \frac{\pi}{2} \times B \times L$$

$$B = \sqrt[3]{\frac{4 \times V}{(L/B)}} \qquad L = B \times (L/B)$$

V = Displacement of the ship (ft^3)

The friction constant (C_f) is

$$C_f = 0.072 \times (R.N.)^{-0.2} \qquad (23)$$

where

(R.N.) = Reynolds Number

$(R.N.) = 10^5 \times \vartheta_0 \times L$ (Average for water)

Since now the wave drag, the beam, the length, and the ram pressure of each ship are determined, the constant (C) of the wave drag formula is

$$C = \frac{D_w \times s}{(B/L)^2 \times q^2 \times B} \qquad (24)$$

In Figure 4, the result of this calculation is plotted against the displacement of the ships. The figure also shows the theoretically determined values of (C). Figure 4 proves that all measurements are within the theoretical limits and within plus and minus 40 per cent of the theoretically predicted average of (C).

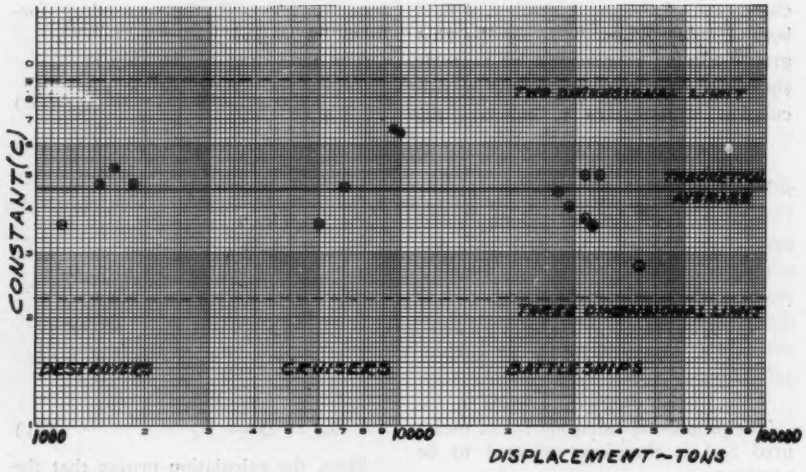


FIG. 4 THE CONSTANT (C) OF THE WAVE DRAG FORMULA VS. THE DISPLACEMENT OF FIFTEEN NAVY SHIPS.

When considering that the fifteen ships selected differ greatly in size, displacement, ratio (L/B), and ram pressure, and that the actual propulsion efficiency is unknown, the result is in favor of the wave drag formula derived. It can even be hoped that more appropriate measurements and some refinements of the approximate theory will lead to a complete agreement between the theory and the measurements.

CONCLUSIONS

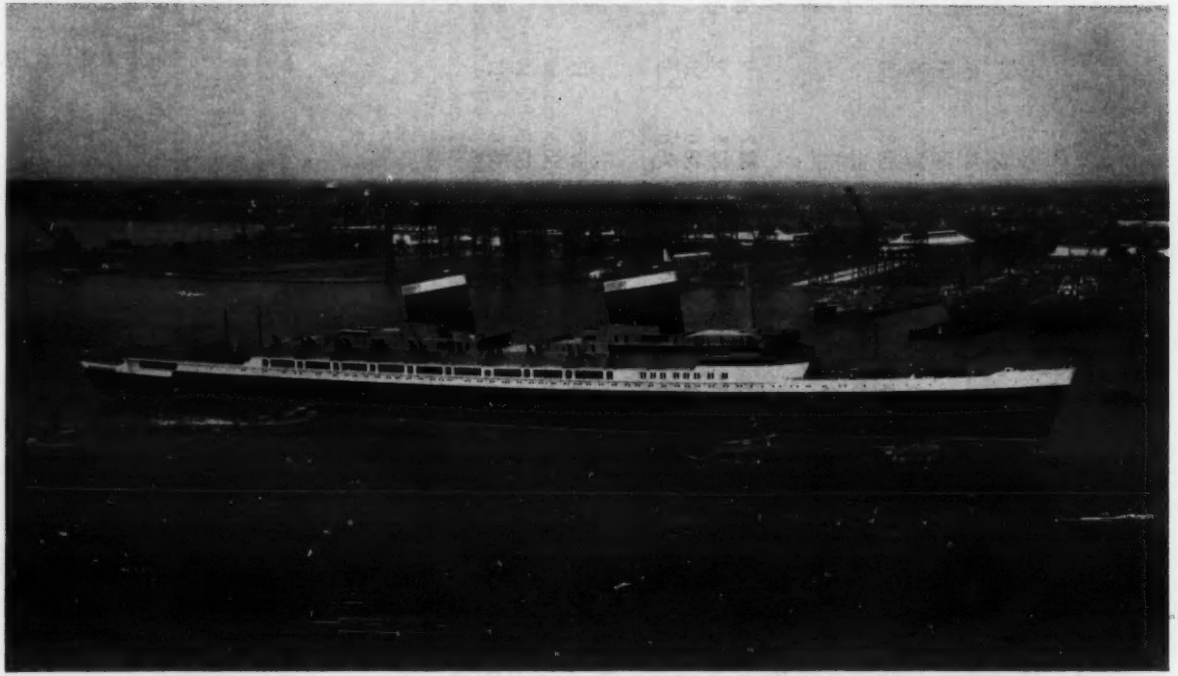
To the best of the author's knowledge, no theoretical expression for the wave drag of ships has been derived which agrees with actual measurements.

The foregoing derives a theoretical wave drag formula which agrees within plus and minus 40 per cent with official data of fifteen ships of the United States Navy (Destroyers, Cruisers, and Battleships).

It can be hoped that more appropriate measurements and some refinements of the approximate theory will lead to a complete agreement between the theory and the measurements.

APPENDIX

Year	*Name*	*Displacement (Tons)*	*Power (H.P.)*	*Speed (Knots)*	*L/B*	*C*
BATTLESHIPS						
1943	*Iowa*	45,000	200,000	33	6	2.7
1921	*Maryland*	32,500	27,300	21	6	3.7
1918	*New Mexico*	33,400	32,000	22	6	3.55
1941	*North Carolina*	35,000	115,000	27	6	4.95
1916	*Oklahoma*	29,000	25,000	20.5	6	4.0
1916	*Pennsylvania*	32,600	33,400	21	6	4.9
1914	*Texas*	27,000	28,100	21	6	4.4
CRUISERS						
1933	*Astoria*	9,950	107,000	32.7	9	6.6
1939	*Wichita*	10,000	100,000	32.5	9	6.5
1942	*Atlanta*	6,000	75,000	38.0	10	3.6
1923	*Omaha*	7,050	90,000	35.0	9	4.5
DESTROYERS						
1940	*Benson*	1,630	50,000	37.0	11	5.1
1937	*Craven*	1,500	42,800	36.5	11	4.7
1937	*Somers*	1,850	52,000	37.5	11	4.65
1916-1917	—	1,190	26,000	35.0	11	3.6



S.S. *United States* Recently Launched by The Newport News Shipbuilding and Drydock Co.

A STATISTICAL APPROACH TO THE MEASUREMENT AND ANALYSIS OF EXPERIMENTAL DATA

NORMAN H. JASPER

THE AUTHOR

The author received his Bachelor's degree in Mechanical Engineering from the College of the City of New York in 1941. He joined the Scientific section of the Puget Sound Naval Shipyard in 1941 and worked there until transferred to the David Taylor Model Basin in 1946 where he has primarily been concerned with applying vibration theory and experimental methods to the study of ship structures. He studied, since 1941 at the University of Washington, the George Washington University, the Graduate School of the Bureau of Standards and the University of Maryland from which he received the Master of Science degree in 1950.

It will be in order to clarify the subject of this paper a little more than is indicated by the title. The term "statistical" in the title may be misleading to some extent since we will not concern ourselves here with the mathematics of statistics but rather will deal with the treatment of data in the aggregate and in doing so will want to extract certain common denominators according to which such data may conveniently be presented.

The prime purpose of this paper is to focus attention on and to indicate an approach to the problem of presenting data, especially experimental scientific data, in a form in which it may readily be put to use by the consumer, often an engineer, whose primary interest is to determine whether a given structure will do the job and who has little time or inclination to go into a detailed analysis of the individual steps that are taken to get the data into a usable form.

The need is indicated for a means of classification of experimental data in some suitable fashion in order to make its engineering use practicable. The time and economic factors entering into the analysis of test data will be touched upon only lightly at this time but they are the prime factors which started me on the present approach. I will illustrate the application of this approach by describing, rather, briefly, several instruments and techniques that have been developed or that are presently under development.

Let us assume that our prime interest lies in experimental analysis of strains,

stresses, and forces. Also, to many, the experimental methods are the means by which analytical theories may stand or fall. The methods and instruments discussed here are applicable to many types of measurements but I shall, for illustration, use the study of the "strength of ships at sea" presently carried out at TMB; the problems encountered there will be, I believe, of interest.

In general, we are given a problem which requires us to determine the action of a body, say a structure, under certain conditions. The action on and the reaction of this body must be specified in terms of quantities that we are familiar with, such as temperature or displacement.

Now there are so many quantities that might possibly affect the action of our "structure" that one must use considerable judgment in his choice so as to permit a practical approach to the solution without, on the other hand, omitting from consideration factors which may have an appreciable influence on the behavior of the structure. In most cases, it is quite apparent that certain factors have but little or no bearing, but it is not as easy to find those which do have significant bearing on the problem.

A convenient method of determining the relative significance of several parameters is to set up a mathematical model, for example, the differential equations that are to represent the physical behavior and determine therefrom the variation of the desired quantity as a function of these parameters. Care must be taken, however, that the mathematical model does, in fact, represent the physical problem to the desired accuracy. Another and more direct method would be to resort to experiment.

It will be found after such a procedure that the action of the structure can in general be specified by several parameters which in themselves are functions of a number of variables.

So the general problem is a rather formidably complex one. The engineer, and I assume other people as well, like to see things remain as simple as practicable. Simple relationship may perhaps be visualized, they are less obscure and, after all, that's what we want—less obscurity.

In the study of the response of structures under load, shock or vibration, many tests are made and a great deal of mathematical and experimental analysis is carried out. In order to effectively present the results of this analytical effort to the engineer and designer, it is necessary to specify in some relatively simple manner the behavior of the structure as a function of some characteristic of the applied force or motion. For example, assume that it has been shown that strain is an index of the probability of structural failure and that the maximum intensity of the applied load can be related directly to the strain, then the salient criteria of the structural behavior for this example are the strain or the maximum load.

The latter illustration describes a very simple relationship. It is readily seen that a somewhat more complex case can be treated similarly. For example, the case where the load is a known function of time, here some characteristic time may need to be added to the criterion of the preceding example. One can go on to more and more complex cases. For engineering use such data can be presented in the form of tables and graphs which give the relationship between the several factors.

Much time and money can and is spent in the search for these significant factors. A criterion may, of course, itself be a function of several others. Once a set of criteria has been established for a particular problem, it will, in general, save a great amount of time and money to design and utilize apparatus which measures the criterion directly rather than the multitude of

factors which originally had to be measured.

In most experimental investigations that I am familiar with, data was measured by instruments which recorded the desired quantities as a function of time and the instrumentation required the more or less continued attention of personnel. This type of test is probably the type most frequently encountered. In some studies it is desirable to base results on sampled data. In this case it is desirable to record the output of a transducer at specified intervals for a given duration of time. It is readily seen that under the latter circumstances it would be advantageous to possess an automatic recording apparatus which would perform the required operation without necessitating human attention.

We may go one step further and require that instead of recording the time function directly, our apparatus also perform a function of data analysis and give us, in some manner, the result of this analysis. The value of this latter approach is readily seen when one considers the relative time and costs involved in a routine analysis of oscillograms. To illustrate, suppose we wish to find the suitability of a structure which is subjected to alternating loads of the type that may cause fatigue failure. We will measure the strain-time history of this structure. The oscillogram thus obtained must, at great expense, be analyzed to determine the magnitude, number and sequence of cyclic strain variations together with the mean level about which these variations occurred. Our automatic instrument would perform this analysis for us and give the result in terms of numbers. A visual analysis of the oscillogram would probably have been prohibitive in most cases, whereas the automatic analyzer has made the final data immediately available at any stage of the test.

We have just mentioned two of the three types of instruments that will be illustrated later on. They are

(1) Apparatus that measures and records the data directly and at the same time performs a statistical sampling operation.

(2) Apparatus that immediately analyzes the signal it receives from the transducer without retaining the original data.

(3) Apparatus that records the signal received from the transducer in a manner such that the original signal or some function thereof may be reproduced at any time permitting laboratory analysis of the recorded signal. A well known example is the magnetic recorder.

It is these three types of instrumentations that we will be concerned with in this paper.

Much could be said about the statistical approach to the collection of data but I will merely make the self-evident statement that much time and money can be conserved by a properly designed statistical investigation. It is assumed here that the mathematicians have supplied us with the proper sampling times and sampling intervals, as well as the accuracy to be expected therefrom. It will, I believe, be found relatively easy to define the latter requirements. It is not necessary that the engineer be trained in statistical mathematics in order to collect data on a statistical basis. He will do well to talk his problem over with persons trained in that field and to let them determine the necessary factors.

I will first give some background information on the problem that really started us on the development of automatic statistical instruments. The Navy's Bureau of Ships requested in 1948 that the David Taylor Model Basin measure the strains and motions of ships at sea for the purpose of evaluating and improving design theories and calculations. Another parallel project required

us to determine the forces on bow structures of ships. This overall study can take in quite a bit of territory. It was necessary to study the overall problem objectively and to decide in the light of the available time and funds upon a general program that could reasonably be accomplished with the available facilities. I have made a preliminary outline of such a program in a paper published in the August '50 issue of the Journal of Naval Engineers entitled "A Statistical Approach to the Longitudinal Strength Design of Ships." As part of the project, it was proposed to collect data on many ships operating on the various sea routes and under a variety of sea conditions. The data to be collected were to give an index to the ships' seaworthiness, the acceleration experienced by the hull girder as a whole and the severity and manner of stresses which the hull must withstand. It would be impractical to secure this mass of data by direct oscillograph recording such as has been done on prior such tests on ships at sea. It was, therefore, decided to develop instrumentation which would obtain the desired data automatically without requiring attending personnel. The objective is to provide an instrument that is installed on the ship and then remains with the ship for a minimum of a complete voyage, say one month, and from which the desired data may be removed at the end of that period. The required instrumentation, in addition to providing the proper information, must be rugged, reliable, and economical in cost and operation.

The first type of apparatus that I will describe is one that is most like the oscillographic equipment that we are familiar with. It is essentially an oscillograph which has provision for automatic sampling and which is designed to operate continuously for weeks on shipboard without necessitating human attention. It is called the "TMB Ships' Motion Recorder" but could be used for purposes other than indicated by its name. It should be remembered that this is the first and only model built and that we will probably wish to add additional features to later models. As built, the recorder operates from a 110 a-c line and is designed to record the ship's rolling and pitching acceleration, its linear acceleration as well as the rolling and pitching angles. It is desired to record only the rather low frequency variation associated with the gross motion of the vessel, that is, rolling, pitching, and heaving motions. This requires that the response to high frequencies be attenuated.

The voltage output of each transducer is amplified and its phase shifted so as to provide maximum power input into one phase of a two-phase motor, the other phase is connected directly across the line. The motor operates a null balance system. The direction and amount of rotation of the motor are directly proportional to the direction and magnitude of the transducer signal. Since a null balance system is used, there is no appreciable error due to lack of voltage regulation in the power supply. The two-phase motor actuates a standard type of pen drive. The pens scribe on a plastic coated recording paper.

The paper drive is actuated by a set of cams as follows:

(a) The paper drive automatically operates and the pens are actuated for a preselected period at regulated intervals, say for two minutes out of every hour.

(b) By manually turning a timer knob, the recorder will record for any desired continuous period up to two hours. At the end of the period the automatic cycling will take over.

(c) The recorder will run continuously if desired.

The paper speed we use is either 1 or ½ inches per minute; 100-foot length of paper is sufficient to permit up to 50 days' operation without reloading the magazine. In use it is intended to select the optimum sampling interval and sampling duration for the particular problem under consideration. The prime advantages of the motion recorder are:

(a) ruggedness of construction and use of subassemblies for ease of replacement and servicing
(b) a clean immediately visible recording trace not requiring processing and not likely to give trouble in operation. The recording paper does not smudge readily.
(c) null balance system
(d) high sensitivity, about 9 m.v. input for full scale deflection
(e) flexibility in application
(f) low cost of installation, operation and low first cost

Additional features can readily be added to the present unit. For example, the recording may be controlled by the magnitude of the quantity to be measured or by some other quantity. A definite limitation of the recorder is its frequency response limitations.

Next I will proceed to an instrument quite unlike the preceding one.

One of the questions that arises in the study of the strength of ships at sea is whether or not fatigue is an important consideration in the strength of ships; also we would like to know the intensity and frequency of occurrence of strains in the ship girder over long periods of time.

For this purpose, we have developed the "TMB Strain Cycle Gage and Counter." This instrument comprises a special 10-inch mechanical strain gage and a box containing the counters and the counting circuit. The counter provides an automatic tabulation of the number of cyclic variations of selected strain amplitudes classified according to the mean strain levels about which these cyclic variations occur. To determine the number of variations about a given mean strain level, it is merely necessary to subtract the reading of one counter from that of another counter.

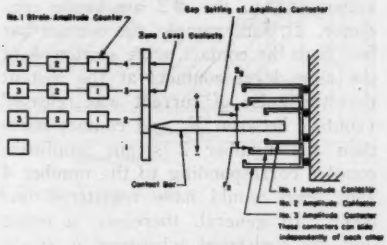


FIGURE 1. Diagrammatic Sketch of Strain Cycle Gage and Counter.

The principle of operation is probably best illustrated by a simplified diagram Figure 1, the main components are (a) a contact bar which is given a linear motion directly proportional to the strain, (b) a series of amplitude contactors which are free to slide as indicated, and (c) a series of zone level contacts which remain stationary. If the contact bar touches the upper contact point of an amplitude contactor and then touches the lower contact point of the same contactor, a counter will count one unit at the moment the contact is made. The particular counter which will register is determined by the level contact which is touched by the contact bar at the moment of counting. Assume that the unit is in the position shown in Figure 1 for zero strain and that the contact bar moves parallel to itself due to an applied tensile strain. The upper contact points "S" of the amplitude contactors #1, #2, and #3 are successively touched by the contact bar. The instant contact is made between the contact bar and the point S_2 of the amplitude contactor #2, a relay in an

electrical circuit associated with this amplitude contactor is cocked. This relay does not close until the strain, and consequently the contact bar, reverses its direction and contact is made at F_2 at which time the relay closes and a pulse of current is sent through the circuit and channeled via one of the zone level contacts to the counter corresponding to the #2 amplitude contactor. If, for example, the contact bar had been in contact with section 4 of the zone level contacts at the instant that the pulse of current was released (contact between F_2 and contact bar), then the number 2 strain amplitude counter corresponding to the number 4 zone level would have registered one count. In general, therefore, a count will be registered whenever a strain variation equal to or greater than that corresponding to a given gap setting of the amplitude contactors has occurred.

For the sake of simplicity, let us eliminate the zone level contacts. Then there will be only one counter per amplitude contactor. Now assume that the mean stress is zero and that there are 1000 cycles of 5000 psi stress; 500 cycles of 10,000 psi stress and 100 cycles of 15,000 psi stress. Then the counter corresponding to the 15,000 psi stress would indicate 100 cycles, that for the 10,000 psi stress 600 cycles, that for the 5000 psi stress 1600 cycles. It is apparent that each counter registers all the strain cycles which are equal or larger in magnitude than that corresponding to that counter. In order to find the number of cycles of strain corresponding to any counter, it is then only necessary to subtract from the reading of that counter the number of counts registered on the counter corresponding to the next larger strain amplitude. If the mean stress had been any value other than zero, the result would not have been changed since the level contact had been eliminated in the arrangement under discussion.

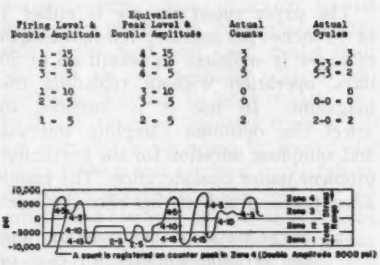


FIGURE 2. Sample Analysis of an Arbitrary Strain Variation.

Now if the zone level contacts are utilized, the analysis of the data given by the counter readings is very nearly the same as that just given except that now there will be a set of counters for each zone level but the analysis for each one of these sets is exactly the same as that outlined before. For purposes of separation of the counts, the counters are arranged according to the zone in which the peak strain occurred, the arrangement is indicated in Figure 2, which also gives a detailed analysis of an arbitrary strain pattern. A careful inspection of this Figure should be sufficient to illustrate the principle of analysis that is utilized.

The physical strain cycle gage will now be described. The gage is illustrated in the sketch, Figure 3, and the photograph, Figure 4. It has a base length of 10 inches and the actual strain is magnified ten times through a lever system. The amplitude contactors are carried by a pivoted and balanced arm which is held in place at any position by a suitable friction pivot. Both the zone level and amplitude contactors are made of monel and are readily adjustable over a wide range. The gage is held to the structure by a predetermined spring force. Two types of gage points have been provided for, conical gage points which require accurate drilling of gage holes or knife edge type gage

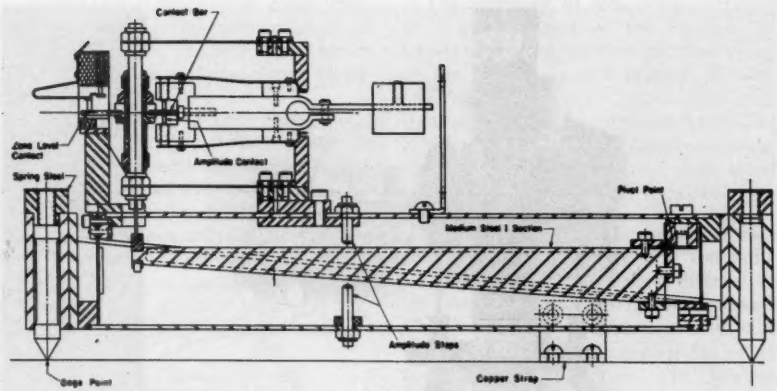


FIGURE 3. Sketch of Strain Cycle Gage.

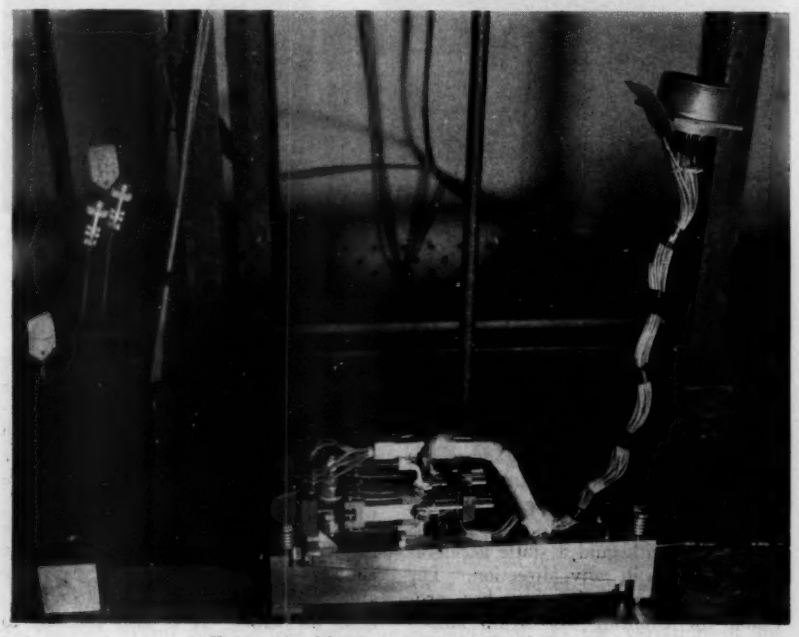


FIGURE 4. Mechanical Strain Cycle Gage.

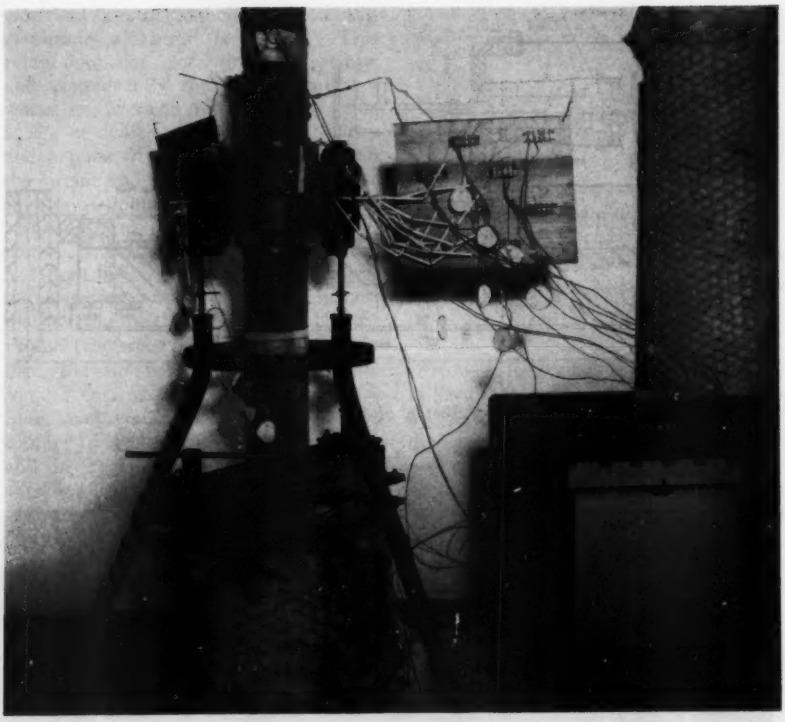


FIGURE 5. Flange assembly in place, showing quick-opening valve, supporting tripod, by-pass line, and recording potentiometer.

points which make for a simpler installation. Rapid temperature changes of the structure were not anticipated. Temperature effects were to be taken care of by (a) making the gage of the same material as the structure and (b) minimizing the temperature lag between the gage and the structure. The gage is designed to withstand a static acceleration of 5 g in any direction. The prototype strain cycle gage has provision for counting six strain amplitudes and classifying them into six zone levels (see Figure 5). The manufacture of the gage requires precision workmanship.

The basic idea of the strain cycle gage can be utilized in other applications where it is desired to count cyclic variations. The general arrangement of such a system could consist of a transducer which transforms a variation of the quantity to be measured into a mechanical motion of a contact bar or of a rotor carrying several contact bars. The contact bars would then actuate the amplitude contactor and counting circuits as before. It is also possible to perform the function of the floating contactor electronically; however, such a system would appear to be unduly complex for our purposes.

Application of this system could probably be found in the cyclic analysis of voltages, currents, pressure, SR-4 strain gages, temperatures, atmospheric conditions, etc.

It is seen that the strain cycle gage and counter provide an automatic statistical analysis of the strain variations in a structure over long periods of time at a great savings of time and money over the conventional time-base data. A rough time base can be provided for the strain cycle data by photographing the counter readings at selected intervals of time.

If it is found that fatigue is not a significant consideration in our particular problem, then it would be unnecessary to go to the trouble of classifying the strain data as before. It may be sufficient to know what percentage of the total operating time the strain falls within given limits. To provide this information, we have developed another instrument which performs a measuring analyzing and totalizing functions. I call this instrument a function totalizer or, in our particular case, a strain totalizer. The essential components are a strain gage, a switching or classifying device and a series of clocks. The operation is as follows: The output of a strain gage is used to change the bias of a vacuum tube. Each vacuum tube will conduct current when its grid voltage exceeds a fixed value which value is in turn set to correspond to a definite strain. Each tube controls a clock which will run as long as the tube is conducting current. Therefore, the total time indicated by any clock indicates the total time that the strain has exceeded the value corresponding to that particular clock. The total time that the strain was in a given region is given by the difference of the clock readings for the two strains defining that region. This data can immediately and conveniently be represented by a simple graph.

The accuracy of this device depends to a great extent of the rapidity of strain variations and on the time delay of the clocks. In one present installation, we will utilize the same strain gage that was part of the strain cycle counter in the strain totalizer. However, almost any electrical or mechanical gage could be adapted to the strain totalizer.

In the preceding two examples, it was seen that the analyzing and totalizing function was accomplished directly at the time that the variation occurred. This, of course, requires that each gage be equipped with the complete analyzing and totalizing apparatus. The question immediately arises: why not have a setup in the laboratory which will do these operations and thus have one set of equipment to analyze the data obtained from a number of transducers. This would not be too great a problem if it were required to record the data over short intervals only; but since we wish to analyze data obtained over a period of weeks or months, conventional recording on magnetic tape or on film would appear to be impractical. We are, at present, devising a method which will make this approach practical; for want of a better name, I shall call it the step-sequential system of recording. The idea which I will presently explain was suggested to use by Mr. G. W. Cook, Chief of the Electronics Engineering Division of TMB.

Essentially the process of recording approximates the signal by a series of steps, the magnitude of each incremental step may be taken as small as desired in order to obtain the required accuracy. For an increase of the measured magnitude by one step, a single positive pulse is impressed on the recording medium, for a decrease of the measured magnitude by one step a single negative pulse is impressed. If the variation of the signal as a function of time is desired, the speed of the recording medium will provide the time base. If only the

sequence of the function but not the time variation is important, then the recording medium need advance only when a variation in the signal must be recorded.

For example, let us assume that we wish to analyze a voltage that varies between zero and 1 volt and that the requirements of the problem are satisfied by approximately the variation in increments of 0.1 volt. This will require ten steps. We will select magnetic tape as one recording medium and every positive increment will be represented by a positive pulse of magnetism and every negative increment by a pulse of negative magnetism. If the voltage is varied from zero to 0.4, then decreased to 0.2 and again increased to 0.9 volt, this sequence will be represented by the following sequence of magnetic pulses, 4 positive, 2 negative, 7 positive.

The actual apparatus can be constructed in a number of ways. The medium may, for example, be magnetic tape, ordinary paper tape or a metal cylinder. The laboratory equipment needed to analyze this data would in the most general case be reproduced so as to approximate the original time function and automatic analysis could then be performed as before. Of course, simpler approaches may be used for special purpose instruments. It is my personal opinion that the approach just outlined may find important applications.

To recapitulate, the instrumentation that has been described is of three distinct types; classifying them according to their function we have:

(1) Apparatus that measures and records the data directly but does at the same time perform a statistical sampling operation—as illustrated by the TMB motion recorder.

(2) Apparatus that immediately analyzes the signal that it receives from a transducer without retaining the received data—as illustrated by the strain analyzer and by the strain totalizer.

(3) Apparatus that records the signal received from the transducer in a manner such that the original signal or some function of the original signal can be reproduced at will, permitting automatic laboratory analysis of the recorded signal; into this clan fall magnetic recorders, photoelectric type of recorders or the step sequential type of recorder.

These systems may be combined in various ways. The method of sampling or the method of analysis may be altered to suit particular conditions. For example, one might arrange to have the data recorded or analyzed only when the measured magnitude exceeds a predetermined level, thus the operating time of the apparatus is reduced.

It has been one objective here to focus attention on the possibilities of utilizing apparatus that will automatically perform the function of sampling, recording and analysis. In this type of approach one is forced to plan the experiment in a form suitable for analysis. It is, of course, desirable to visualize the analysis of data when planning any type of experiment but in many cases one unfortunately does not know what the significant factors are. It is believed that some programs of study that would be impractical of accomplishment if attempted in the usual manner will become feasible if approached along the lines suggested here, due primarily to the great savings in man hours and the almost immediate availability of significant results.

NOTE: A patent covering the basic features of the strain cycle gage was issued to R. T. McGoldrick and L. B. Tuckerman (U.S. Pat. 2378422).

PART TWO

REPRINTED ARTICLES

The articles contained in this Part Two represent those which, in the opinion of the editorial staff, are the most valuable which have been gleaned from recent issues of contemporary publications.

This material was formerly published under the heading "NOTES". It has been given somewhat more prominence and made more readable to enhance its value. The editors' purpose here is to provide a convenient repository to readers of the JOURNAL, for articles originally published elsewhere which contribute to the purpose of the Society: to further the advancement of naval engineering.

MACROGEOMETRIC CHANGES OF CYLINDER FORM, THEIR NATURE AND EFFECT ON THE OPERATION OF INTERNAL COMBUSTION ENGINES

(VESTNIK ACADEMII NAUK—JUNE 1950 P. 130)

TRANSLATED FROM THE RUSSIAN BY MR. E. C. MAGDEBURGER

A dissertation under the above title was presented by Sh. M. Bilik to the Machinery Institute in Moscow, USSR in seeking a doctors degree. It consists of two volumes and six chapters.

In the first chapter the author justifies the importance of his theme to the national economy and emphasizes the role of internal combustion engines in industry and agriculture. Pointing out that the cylinder group, i.e., liner and rings, constitute the weakest link limiting the time between overhauls the author bases his deductions on the service data of automobiles. The author concludes that the current insistence on the highest finishes for the bores of cylinders overshadows the important effect of accuracy of dimensions and form. Although various engine builders have established that cylinder deformation has considerable influence on the operation of it, this very important topic has not been studied sufficiently and deserves more attention.

In the second chapter are given results of actual measurements of the changes in the geometry of the cylinder during the process of assembly and these cylinder deformations are carefully analyzed in detail. To evaluate these cylinder deformations the author set up the following parameters: maximum ovality, ovality in two perpendicular planes, conicity of sections, maximum conicity and oval conicity.

The third chapter created the most interest among his official opponents. The author has actually measured the deformation of many thin-walled tubes of various geometric dimensions when subjected to different internal pressures. In addition to the measurements of deformation the distribution of stresses was also determined by brittle lacquer method and tensiometers. As a result of this investigation the author established the relationship between the character of deformation and the geometric parameters of the tubular body and adopted such concepts as influences of ends, linear and pseudo-linear effects.

In the fourth chapter the author discusses the changes in the macrogeometry of the cylinder as influenced by temperature and length of service. For this purpose the author conducted special investigations on installations simulat-

ing the warming up of the cylinder group under stationary conditions. Based on these investigations it was possible to establish the laws governing the changes of cylinder form when subjected to heating, and as a general rule heating increases the distortions resulting from assembly. The effect of service on the macrogeometry of liners was determined from the results of testing single cylinder engines. As the result of macromeasurements during the various stages of these prolonged investigations the author came to the conclusion that it is possible by introducing definite macrogeometric changes into the original cylindrical form of liners to localize to a greater or lesser degree the effect of distortions of form resulting from either deformation or wear.

The fifth chapter is devoted to the evaluations of the effect of macrogeometry of cylinder on the operation of the engine—by measurement of gas blowouts, lubricating oil consumption and wear of the cylinder group. The author used two specially prepared installations to investigate blowout of gases from the combustion chamber into the crankcase for various deformations as a function of pressure variation, position of piston, type, distribution and number of piston rings.

Investigation of wear of contacting surfaces of the cylinder groups during operation of the engine permitted the author to advance several concrete recommendations to lengthen the service life of engine cylinders and consequently of engines themselves.

The last or sixth chapter summarizes all deductions and recommendations covering the whole subject matter that may be applicable for future designing and maintenance of the cylinder group of internal combustion engines in the transport field.

One of the official opponents commended the author for his brilliant and original scientific investigation pioneering the problem of macrogeometric deformations of the cylinder group of the internal combustion engines in the transport field. Careful theoretical analysis of the results of the extensive experimentation and tests by the author developed into an original theory of the phenomenon of linear deformation of tubes loaded on their ends first discovered by him.

The investigations practical usefulness stems from the fact that it permits the designer to incorporate concrete means to overcome the difficulties experienced in properly wearing-in the cylinders of internal combustion engines.

The carefully worked over results of Mr. Bilik's extensive experiments supply the basis for qualitative and quantitative criteria for establishing conditions of macrogeometry of coworking surfaces of the cylinder group.

FACTORS AFFECTING SUPERHEAT CONTROL

ACKNOWLEDGMENT

This article, by MARTIN FRISCH, Vice-President in Charge of Engineering, Foster Wheeler Corporation, was presented at a meeting of the Detroit Section of The American Society of Mechanical Engineers in 1941. It was originally published in "Mechanical Engineering," September 1941, and republished, with slight revisions, in the December 1950 issue of "Heat Engineering." The paper outlines the various means now in use for controlling the temperature of superheated steam from a steam generator, and discusses the factors influencing superheater life and reliability. Although the article applies mainly to coal-fired stationary boilers, the design points exemplified are equally applicable to marine boiler designs.

It is proposed in this article to outline generally the most important factors affecting superheat control and to describe some of the means used for this purpose with their relative advantages and disadvantages. Since superheat-control problems at high steam temperatures have accentuated effects tending toward superheater deterioration, it is also proposed to discuss factors affecting superheater life and reliability.

The types of control to be discussed are those employing the following:

1. By-pass and damper.
2. Desuperheater; evaporative type.
3. Desuperheater; feedwater-heating type.
4. Condenser humidifier.
5. Combination superheater; conventional.
6. Combination superheater; separately fired type.

When generating steam at high pressure and temperature with slag-forming fuels and a convection superheater, precise steam-temperature control over a wide load range is possible only by compromises in design which impair the reliability of the steam generator as a whole. The size of a convection superheater depends, among other things, upon the minimum load at which the full specified temperature is required, the gas temperature, rate and type of gas flow available at the superheater, and the feedwater temperature.

If the amount of furnace cooling surface installed ahead of the superheater is large enough to reduce the furnace gas exit temperature sufficiently to freeze all entrained ash particles before reaching the superheater, and if the superheater tubes are spaced on wide centers to prevent bridging, the required final steam temperature may not be attainable over the desired load range with any superheater of practical size. As the gas temperature to the superheater decreases, and the load range over which it is desired to maintain a given steam temperature constant increases, the size of the super-

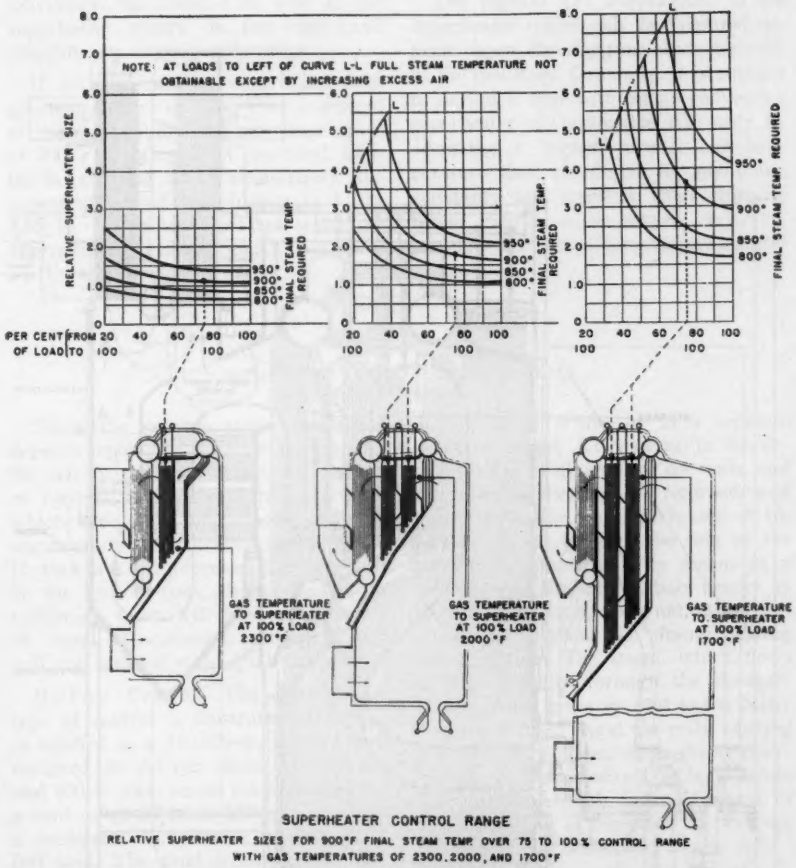

NOTE: AT LOADS TO LEFT OF CURVE L-L FULL STEAM TEMPERATURE NOT OBTAINABLE EXCEPT BY INCREASING EXCESS AIR
RELATIVE SUPERHEATER SIZE
FINAL STEAM TEMP. REQUIRED
950°
900°
850°
800°
PER CENT OF LOAD FROM 20 40 60 80 100
TO 100 100 100
GAS TEMPERATURE TO SUPERHEATER AT 100% LOAD 2300 °F
GAS TEMPERATURE TO SUPERHEATER AT 100% LOAD 2000 °F
GAS TEMPERATURE TO SUPERHEATER AT 100% LOAD 1700 °F
SUPERHEATER CONTROL RANGE
RELATIVE SUPERHEATER SIZES FOR 900°F FINAL STEAM TEMP OVER 75 TO 100% CONTROL RANGE WITH GAS TEMPERATURES OF 2300, 2000, AND 1700°F

FIGURE 1. Effect of gas temperature and control range on superheater size. (Pressure 1325 psi at superheater outlet. Conventional wet-bottom pulverized coal fired unit.)

heater increases rapidly. This is illustrated in Fig. 1 based on a study of superheaters for operation at 1325 psi, from which it may be deduced, for example, that if a final steam temperature of 900 F is to be maintained constant from three-quarters to full load, the superheater sizes will vary approximately as 1.1 to 1.75 to 3.65 as the superheater gas-inlet temperature at full load changes from 2300 to 2000 to 1700 F, respectively, superheaters for each condition having the same spacing, gas-mass velocity, and feedwater temperature. Actual surfaces for 300,000 lb per hr would be approximately 3770, 6000, and 12,500 sq ft, respectively. The three diagrams of Fig. 1 illustrate to scale relative proportions of the furnaces and superheaters for these three conditions.

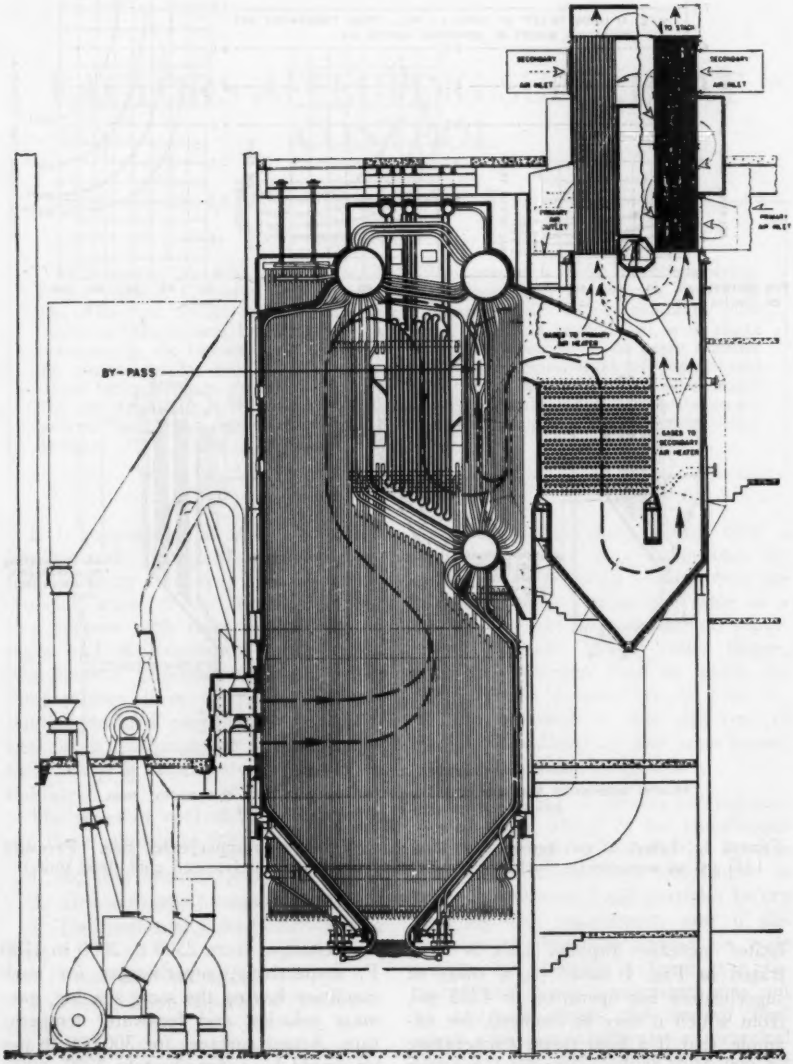


FIGURE 2. Gas by-pass control applied to 550,000 lb. unit. (Working pressure 820 psi, final steam temperature 900 F, from 66 to 100 per cent load.)

Obviously, the furnace as well as the superheater shown on the right-hand diagram are impractically large.

If with a full-load superheater-inlet gas temperature of 2000 F it is desired to maintain a final steam temperature of 900 F constant from one-third, two-thirds, and full loads, respectively, the superheaters will vary in size as 3.5 to 1.85 to 1.6. Actual surfaces would be approximately 12,000, 6350, and 5500 sq ft, respectively.

The highest gas temperature to the superheater which may be tolerated depends upon the slagging characteristics of the fuel fired. Generally, it is possible to design a unit with an all-convection superheater of reasonable size only by accepting a higher furnace-exit temperature than the desire for maximum reliability and freedom from slagging would indicate to be proper, or by reducing the steam-temperature-control range.

METHODS OF CONTROLLING SUPERHEAT

Since the control range obtainable depends upon the furnace design and the size of the superheater, the method of control affects only the ease with which the steam temperature may be regulated to suit turbine requirements. If they are of adequate size, controls of the gas by-pass, desuperheater, or condenser types will be approximately of equal effectiveness, although they will not respond with equal rapidity.

By-Pass Control. The gas by-pass type of control is illustrated in Fig. 2, as applied to a 550,000-lb. per hr. unit, designed to deliver steam at 820 psi, and 900 F. final steam temperature over a load range of 66 to 100 per cent with a feedwater temperature of 300 F. at full load. The final steam temperature is maintained at the desired value by regulating the amount of gas flowing over the superheater by means of a by-pass damper.

Figure 3 shows the application of controls of the desuperheater and condenser types to the same boiler.

Desuperheater - Evaporator Control. In the desuperheater type of control the final steam temperature is regulated by varying the temperature of the steam at an intermediate point in the superheater by desuperheating all or part of the steam in desuperheating coils in one of the boiler drums or in a separate pressure vessel. The steam to be desuperheated is diverted to the coils, and after desuperheating is returned and mixed with the undesuperheated steam before passing through the rest of the superheater. Control is by means of a valve within the intermediate header as shown, or by using external valves and piping to regulate the desuperheating and remixing. The steam, which flows at high velocity through the desuperheating coils, gives up heat to the boiler water which surround the coils, causing it to boil. Sometimes, when the desuperheater coils are located in a separate pressure vessel, the desired amount of desuperheating is regulated by varying the amount of immersion of the coils.

Desuperheater - Feedwater - Heater Control. Another method of controlling by desuperheating is to pass part of the feedwater through coils installed within the intermediate header of the superheater, or by spraying of feedwater over coils through which the steam is passed, located in a separate vessel.

Condenser Control. The condenser type of regulator operates by controlling the humidity of the steam entering the superheater by condensing part of the steam after complete purification by

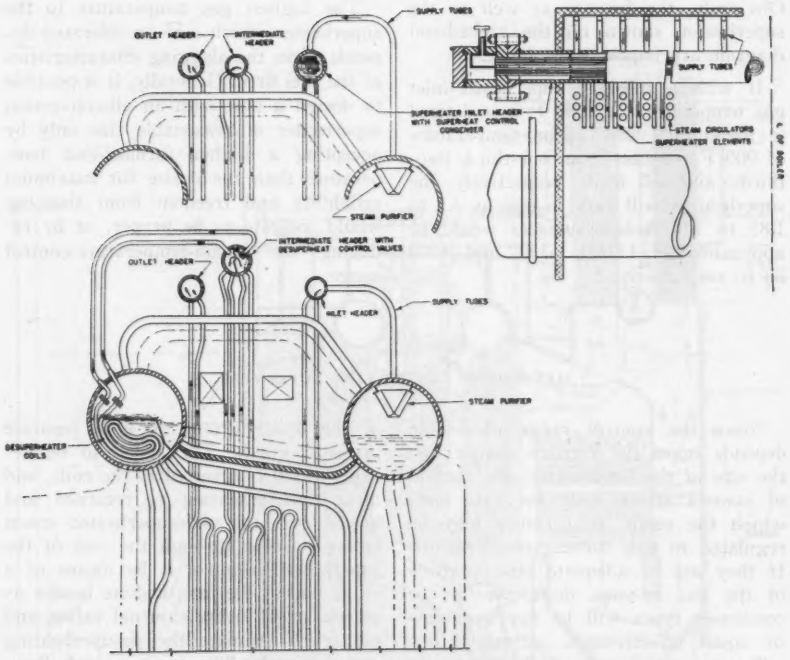


FIGURE 3. Comparison of condenser and evaporative desuperheater control systems for unit shown in Fig. 2.

washing or other adequate means in the boiler, so that the steam is humidified with chemically pure water. This method precludes the possibility of introducing scale-forming solids into the superheater, as in the past with water injection from an outside source. This is accomplished as shown in Fig. 4 by controlling the amount of feedwater flowing through the condenser heat exchanger, located within the superheater inlet header. This method is more rapid than either the gas by-pass or desuperheating methods because it requires less surface for the same control range than desuperheating methods, and the temperature of less metal has to be changed when the steam temperature changes. The control valves, operating as they do on water, are relatively small, compared to the large steam valves required in the desuperheater control system for passing superheated steam with low pressure loss. Hence the regulating mechanism is simpler and lighter than required for either by-pass or desuperheater types. The condenser method has the further advantage that loss in pressure of the steam flowing over the condenser surface is negligible, as compared with the necessary pressure loss through or over desuperheating surface, if the amount of desuperheating surface is not to be prohibitive.

The amount of steam condensed for a given range of control in the condenser system is very nearly the same as the amount of water boiled in the evaporator type of desuperheater control.

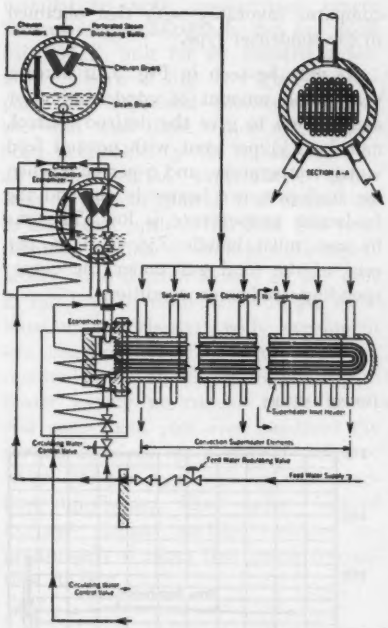


FIGURE 4. Condenser steam temperature control system.

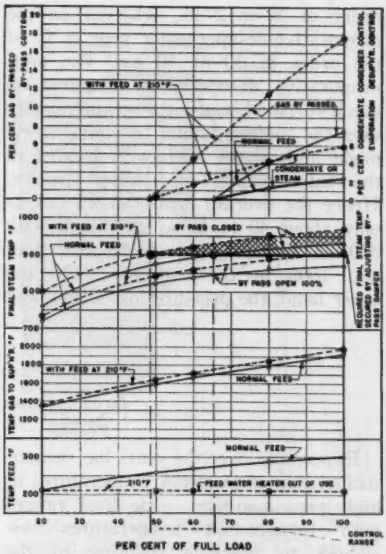


FIGURE 5. Comparison of characteristics, by-pass, condenser, and evaporative desuperheat control systems for unit shown in Fig. 2.

CHARACTERISTICS OF BY-PASS, DESUPERHEATER, AND CONDENSER CONTROLS COMPARED

Figure 5 compares the duty of the by-pass, desuperheater, and condenser types of control on the 550,000-lb. per hr. unit, shown in Fig. 2, in order to maintain a constant steam temperature both with varying load and temperature of feedwater. As the feedwater temperature drops, more fuel has to be burned to generate the same weight of steam per hour, hence the temperature of the gas at the superheater and the final steam temperature both rise. The gas by-pass and damper, as well as the evaporator type of desuperheater, must be sufficiently large to compensate for the effect of decreased feedwater temperature when feedwater heaters are not in service. This may roughly double the capacity of the required by-pass and desuperheater. With controls of the condenser- and nonevaporative-desuperheater types, the reduced feedwater temperature automatically increases the condensing or cooling capacity and compensates to a large degree for the increase in duty due to decrease in feedwater temperature.

The comparative surfaces required are 130 sq. ft. of condensing-control surface, 300 sq. ft. of evaporative-desuperheater surface, and 1400 sq. ft. of nonevaporative feedwater-desuperheating surface. The corresponding pressure losses on the steam side of the condenser- and nonevaporative-desuperheater types are negligible, being of the

order of 1 or 2 lb., whereas on the evaporative-desuperheater type it is of the order of 30 or 40 psi. The non-evaporative desuperheater requires so much surface because the steam velocity over the surface cannot be conveniently made high if the surface is located in the intermediate superheater header. Hence the over-all heat-transfer rate is only about 40 to 50 as compared with 300 in the evaporative desuperheater and 1000 in the condenser. On the other hand, the pressure loss is low and compares favorably with that obtained in the condenser type.

As may be seen in Fig. 5, it takes a very small amount of condensation or evaporation to give the desired control, namely, 2½ per cent with normal feed water temperature, and 6 per cent when the high-pressure heater is out and the feedwater temperature is low. The gas by-pass must handle 7½ to 17½ per cent of the total gas under the corresponding feedwater conditions.

COMBINATION SUPERHEATERS

Experience over the years has demonstrated that high steam temperatures at high pressures over wide load ranges with furnace-exit temperatures low enough to preclude slagging of the superheater may be obtained by the use of combination radiant-and-convection superheaters. It is well-known that as the load increases the steam temperature from a radiant superheater falls, while that from a convection superheater rises. By proportioning the surfaces properly, it is possible to balance these complementary characteristics of the two types of superheaters so as to obtain a final steam temperature which remains practically constant over the entire load range. The deviation from constancy is so small that only minor correction is required. This is illustrated in Fig. 6, which shows actual results on a combination superheater for an installation of moderate pressure and capacity. Control equipment in such installations is required only to correct for changes in feedwater temperatures from the normal. Changes in CO_2, as may be seen in Fig. 6, only slightly affect the final steam temperature of the combination, because though individually the effects of radiant and convection sections are large, the effects are opposite and cancel each other.

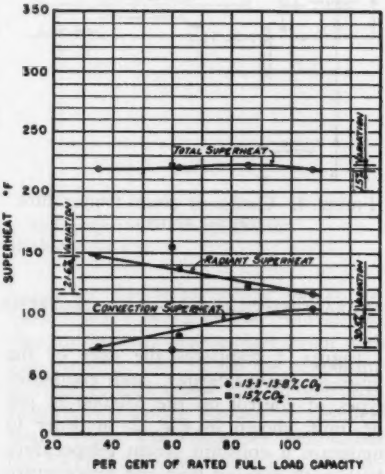


FIGURE 6. Characteristics of combination superheater, 110,000 lb. per hr. unit.

A number of high-pressure, high-temperature installations utilizing combination superheaters in conventional single-furnace as well as in twin-furnace units have been made, based on satisfactory experience over long periods in large and small installations. Typical of many conventional single-furnace-type

installations with combination superheaters is the 300,000-lb. per hr., 1550 psi, 960 F. unit for an Eastern public utility shown in Fig. 7. Figure 8 compares the superheat characteristics of this unit with those which might have been obtainable by the substitution of an all-convection superheater, either with or without control. It is interesting to note that with a full load gas temperature of 2000 F. entering the superheater control from below, 30 per cent to full load is obtainable with the combination superheater with one-fourth less convection surface than would be required with an all-convection superheater having no control range at all and with about just over one-third the surface required for a control range of 45 to 100 per cent with an all-convection superheater. Each square foot of exposed radiant surface replaces approximately 6 times that much convection surface.

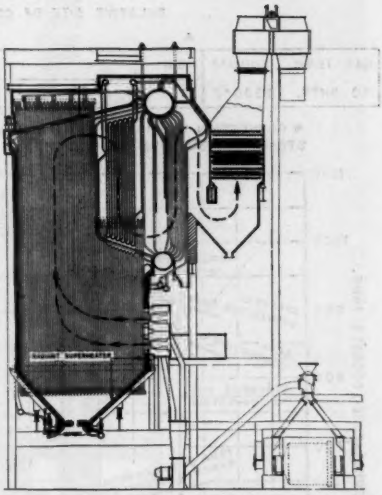


FIGURE 7. Typical combination superheater applied to single-furnace boiler. (Steam output 300,000 lb. per hr. at 1325 psi and 960 F final steam temperature, from 30 to 100 per cent load.)

TWIN-FURNACE BOILERS WITH SEPARATELY FIRED SUPERHEATERS

For yet wider control ranges, and certain freedom from slagging and for precise steam-temperature control, irrespective of feedwater temperature, fuel, or excess air, twin-furnace boilers consisting of a separately fired superheater furnace in parallel with an otherwise similar boiler furnace have found application in the last 15 years. They have been used in situations where rigid steam temperature control requirements had to be met under all conditions with good or bad fuel. Some of these units, ranging in size from 40,000 to 750,000 lb. per hr., have been in marine and stationary service since 1936. Numerous additional marine units of this type for capacities of over 100,000 lb. per hr. and large stationary installations for capacities of 450,000 to 600,000 lb. per hr. with final steam temperatures of 910 to 950 F., maintainable over load ranges of about 6 to 1, have also been built. The stationary units, of which Fig. 9 is typical, were designed with much lower gas furnace-exit temperatures than would be practical with conventional single-furnace units having convection superheaters for the same range of steam temperature control.

The superheater side consists of a completely water-cooled furnace except for the inside wall, which is completely covered by the radiant section of a combination superheater. The radiant-convection sections of the superheater form integral elements requiring no interconnecting piping. Alternate tubes of the radiant section are connected to the steam drum at the top and take saturated steam from it after complete purification within the drum. The steam flows downward to a return header at the bottom, then upward through the adjacent radiant-superheater tubes, their integral convection elements, and to the

RELATIVE SIZE OF CONVECTION SUPERHEATER

	A	B	C
GAS TEMP. 2000°F TO SHTR. 2300°F	1.6● / 1.1	4.4 / 1.9	2.2 / 1.5

● CONVECTION SECTION OF COMBINATION SUPERHEATER ALONE WOULD ONLY RAISE THE STEAM TEMPERATURE TO 900° F

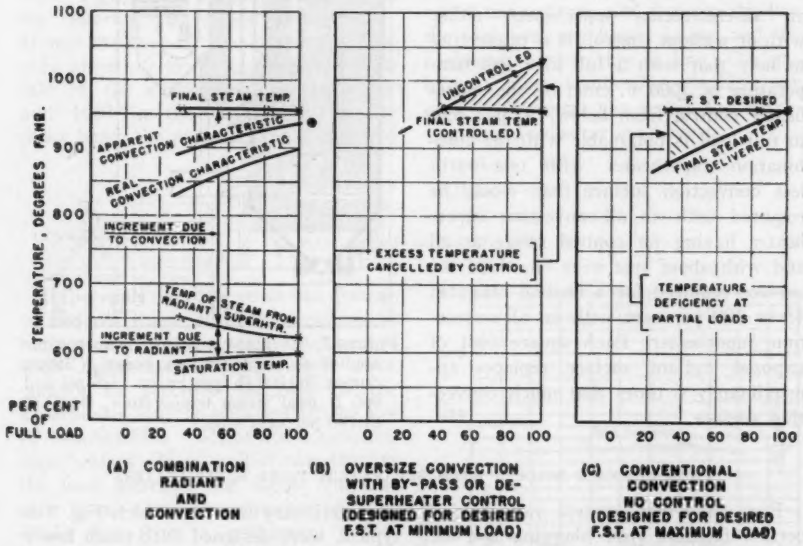


FIGURE 8. Comparison of combination and convection superheater characteristics for unit shown in Fig. 7.

outlet header. The radiant and convection heat-absorbing surfaces are so proportioned as to produce a final steam temperature which deviates insignificantly from the desired value over the entire load range. The small deviation is corrected by adjusting the rate of firing of the superheater furnace. The final steam temperature depends only upon the steam flow through the superheater and the firing rate in the superheater furnace. Hence the final steam temperature may be altered at will by firing more or less fuel in this furnace. Fuel is fired at the top with short-flame burners. Gases pass downward and out of the furnace through an ash screen near the bottom into the convection superheater. After passing through the superheater they flow upward into the economizer. Saturated steam alone may be produced, if desired, by firing the boiler furnace and not firing the superheater furnace.

The boiler furnace is like the superheater furnace, except that the inside wall is a waterwall and the gases, after passing out of the furnace through the ash screen, flow upward through steam-generating surface instead of the superheating surface. The superheater side and the boiler side are separated by a

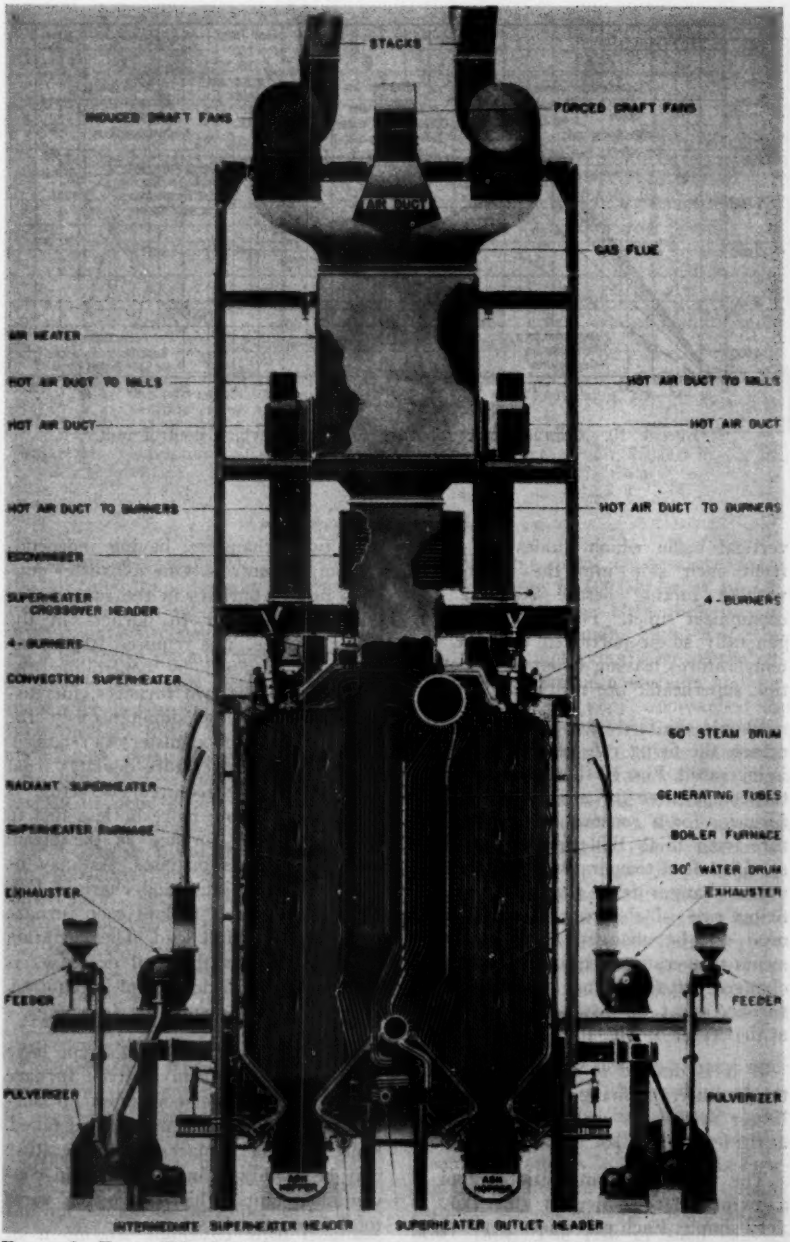


FIGURE 9. Foster Wheeler twin furnace superheat control boiler. (Capacity 600,000 lb. per hr., 900 F final steam temperature at 900 psi, 15 to 100 per cent load.)

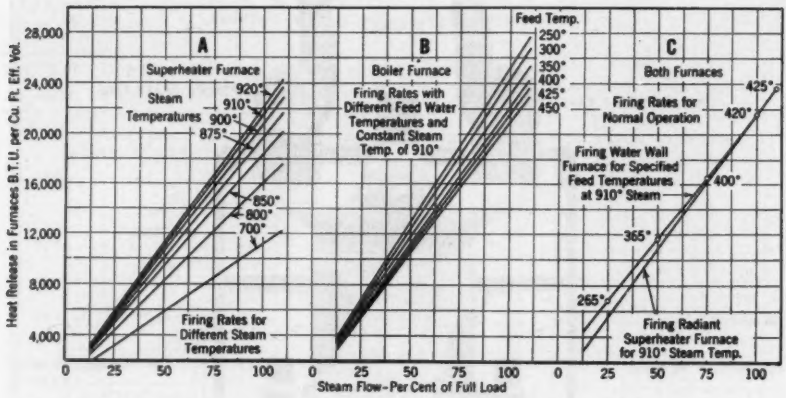


FIGURE 10. Characteristics of twin furnace superheat control unit.

vertical baffle which guides the gases from each side into the economizer without mixing before reaching the economizer inlet. The boiler side is generally so proportioned that the gas temperatures leaving boiler and convection superheater are nearly the same.

Should the feedwater temperature increase, the firing rate of the boiler only is increased. Fig. 10 shows, on the right, the comparative firing rates of the two furnaces for a constant steam temperature from 15 to 100 per cent load with specified feed temperatures. If the feedwater changes from that specified, the firing rate of the superheater furnace need not be changed as long as the steam temperature required remains unchanged, but the firing rate of the boiler furnace must be changed as indicated at the center of Fig. 10.

If it is desired to change the steam temperature, the firing rate of the superheater furnace is changed as indicated at the left of Fig. 10.

The manual or automatic control of a twin-furnace boiler of this type is very simple. Each pulverizer is provided with two exhausters, having independent output controls. One exhauster supplies fuel to burners in the superheater furnace, the other to burners in the boiler furnace. The required steam output is maintained by controlling the total fuel to the two furnaces, increasing or decreasing simultaneously the output of the two exhausters as required. If the steam temperature deviates from that desired, the required correction is made by diverting more or less fuel to the superheater furnace as the steam temperature falls or rises. As may be seen from the right-hand chart in Fig. 10, the amount of fuel to each furnace is the same at full load. The deviation from equality as the load decreases is small and very little control is required. Control may be manual or automatic.

Twin-furnace units of this type have the advantage that in a given furnace volume 20 to 50 per cent more furnace-wall cooling surface may be installed, hence a lower furnace-exit temperature may be readily maintained than in conventional units which have the same total furnace volume.

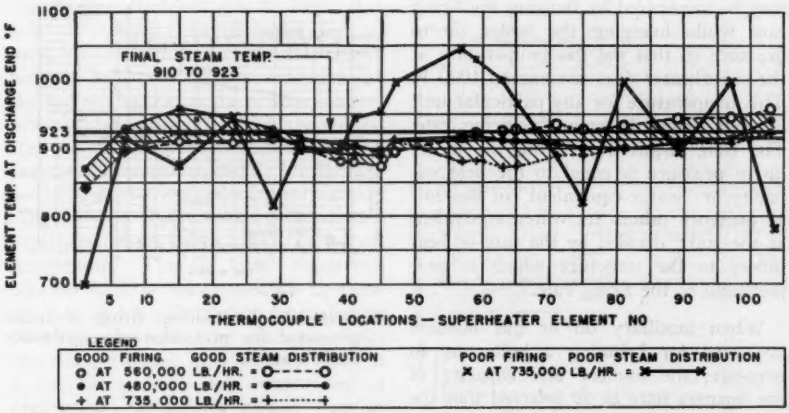


FIGURE 11. Convection superheater temperatures. (Twin-furnace unit, 750,000 lb. per hr., 925 F final steam temperature, 30 to 100 per cent load, at 1325 psi.)

PROTECTION OF SUPERHEATERS

The life of a superheater, whether of the all-convection, radiant or combination type, assuming a steam supply of utmost purity, depends upon the care used in its design and operation.

Superheater tubes may be damaged in service because of inadequate steam flow through them, as a result of the superheater being designed for a pressure drop that is too low. This often results in poor steam distribution between the elements, especially at low output rates. Local high-temperature gas concentrations in portions of the superheater, due to poor firing conditions in the furnace or partial slagging of the superheater, may cause excessively high localized metal temperatures.

Figure 11 summarizes temperature measurements made on the outlet ends of elements of a large convection superheater of the pendent type with good as well as bad gas and steam distribution. This superheater installed in a 750,000-lb. per hr. unit delivers steam at 1325 psi and 925 F. The elements in the hot section of the superheater are 4 to 6 chrome stabilized with titanium. The limiting metal temperature for 1 per cent creep in 100,000 hr. is 1075 F. The steam flow per element in this installation is unusually high. Nevertheless, the steam leaving some of the elements, when firing conditions and gas distribution were poor, approached the creep-limit temperature, indicating that metal temperatures in these particular elements probably exceeded the creep limit. With good firing conditions temperature variations from the mean were reasonably low with a comfortable margin below the creep limit. Experience indicates that past tendencies limiting superheater pressure drop to low values were unwise. Experienced operators now readily accept superheater pressure drops of 75 to 100 psi and even higher for temperatures above 850 F., when control ranges are wide.

Even a properly designed superheater may be damaged if the starting procedure is carelessly handled. Probably more superheaters are damaged while a steam generator is being placed in operation when little or no steam is flowing through the superheater than at any other time. The risk of such damage

may be minimized by limiting the firing rate while bringing the boiler up to pressure so that the gas temperature at the superheater does not exceed 1000 F. This temperature for any particular unit depends primarily upon the firing rate. The time required to bring the boiler up to pressure is equal to the heat capacity or "water equivalent" of the unit at pressure minus its water equivalent at the start, divided by the rate of heat inflow to the structure which is proportional to the firing rate.

When auxiliary oil or gas burners are used for bringing a boiler up to pressure, the number and capacity of the burners may be so selected that the burners may be lit and kept firing continuously at the required rate until the unit is up to pressure. However, if the unit is "direct-fired" with pulverized coal, the minimum stable pulverizer and burner capacity may exceed the firing rate permissible if the limiting gas temperature at the superheater is not to be exceeded. The unit must then be brought up to pressure more carefully by firing intermittently, as shown in Fig. 12, which is typical of the unit shown in Fig. 2. The length of the on and off periods is so selected that the proper total time as previously defined is taken to bring the unit up to pressure. Temperature equalization takes place during off periods throughout the unit, minimizing the effects of possible inequalities in heat absorption of adjacent parts and resultant damaging thermal stresses while circulation is being established uniformly in waterwalls and steam-generating tubes. If adequate vent and drain valves are provided and used on the superheater, some steam flow through the elements will be established as soon as the steam pressure in the unit exceeds atmospheric pressure. This flow progressively increases as the pressure increases.

Sometime the vents are connected to a condenser which not only accelerates

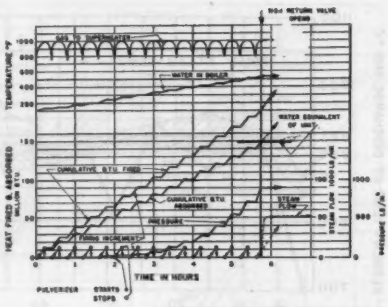


FIGURE 12. Intermittent firing of steam generator for protection of superheater while starting.

the steam flow through the superheater, thus increasing the protection, but serves as a means of preventing condensate loss. When the condenser type of superheat control is used, the condenser provides a means of keeping the superheater flooded with pure condensate during the firing-up process, assuring certain protection for the superheater. Flow of condensate through the superheater is controlled by means of a vent valve on the outlet header.

Radiant superheaters in conventional single-furnace units are favored when starting by bringing the unit up to pressure with the burners most distant from the radiant superheater. With adequate venting facilities, and particularly if the condenser type of control is used and the radiant superheater can be kept flooded with condensate, adequate protection may be obtained until the unit is up to pressure. In twin-furnace boilers superheaters cannot be damaged when starting because no fuel is fired in the superheater furnace until the unit is up to pressure and steam is flowing through the superheater.

Properly designed radiant superheaters are as safe from damage in normal operation as properly designed convection superheaters. As a rule, maximum metal temperatures in radiant-

superheater elements can be kept at a safer low level than in outlet legs of most high-temperature convection superheaters, because it is customary to use the radiant section as the primary superheating stage through which the coldest steam is passed. With a reasonably high pressure drop and adequate steam flow per element over the operating-load range, it is possible to keep metal temperatures well below the creep-limit temperature. This is quite important, because it is no more possible to have the rate of heat absorption by radiation uniform over all the radiant heat-absorbing surface in a furnace than it is possible to have each square foot of convection superheater absorb heat at exactly the same rate. Therefore, it is the maximum, not the mean, metal temperature which determines the safety of any heat-absorbing surface.

Figure 13 shows how the element outlet temperature varies along a radiant superheater of the 750,000-lb. per hr. high-pressure twin-furnace unit already referred to. Corresponding furnace gas-temperature measurements serve as an index to these variations. The steam flow through the elements of this superheater is sufficiently high so that the maximum metal temperature, even in the hottest elements, is well below the creep-limit temperature at all loads. These elements are made of stabilized 4 to 6 chrome.

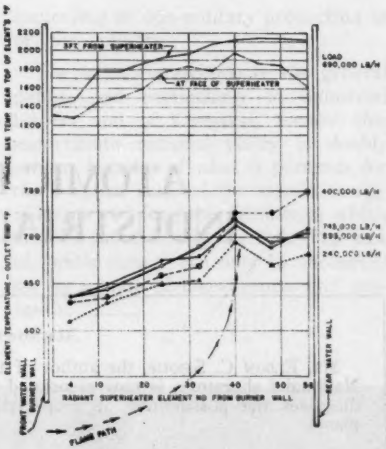


FIGURE 13. Radiant superheater temperatures. (Twin-furnace unit, 750,000 lb. per hr., 925 F final steam temperature at 1325 psi, 30 to 100 per cent load.)

SUMMARY

A high steam temperature may be maintained constant over a wide load range with a convection superheater of reasonable size only by tolerating a high furnace gas-exit temperature.

If high furnace-exit temperatures cannot be tolerated because slag-forming fuels are to be burned, a reduction in the control range of the convection superheater must be accepted.

If neither a high furnace-exit temperature nor a limited control range can be tolerated, it is advisable to use a combination radiant-and-convection superheater, preferably in a twin-furnace type of unit with the combination superheater separately fired.

For regulating steam-temperature controls of the by-pass, desuperheater, or condenser types may be used for conventional superheater installations. Units of the twin-furnace type with separately fired superheaters are controlled by differential firing and such units can be designed for lower furnace-exit temperatures and wider control ranges than conventional units.

Damage to superheaters in operation and during starting up may be avoided by proper care in equalizing gas and steam distribution to the superheater and by proper manipulation of the firing equipment and superheater when starting.

ATOMIC ENERGY—INDUSTRIAL PARADOX

ACKNOWLEDGMENT

MR. ELDON C. SHOUP, the author of this article, former Director, Brookhaven National Laboratory, is now associated with Cresap, McCormick and Paget. He discusses the possibilities of progressive private ownership of atomic energy plants.

Strangely enough it is not the scientific complexities of atomic energy but the problems of economic organization which are proving more than a match for our American ingenuity.

We pride ourselves in this country on having organizational know-how, but we have failed signally to organize the business side of the atom. We are spending billions willingly at home and over the world to preserve, among our other freedoms, our free enterprise system. Yet in this sixth year after Hiroshima we are still shaping this great new industry as a tight government monopoly. Almost heedlessly, and with the force of duly appropriated public money, we are planting the roots of atomic energy firmly in the soil of state socialism.

Any less categorical statement than this of the existing situation would simply ignore the facts. It should be emphasized at the outset, however, that under existing circumstances, notably the restrictions imposed by the Atomic Energy Act of 1946, the Atomic Energy Commission has done what it could to encourage management participation by industry in the program of atomic energy development. New legislation is needed without delay.

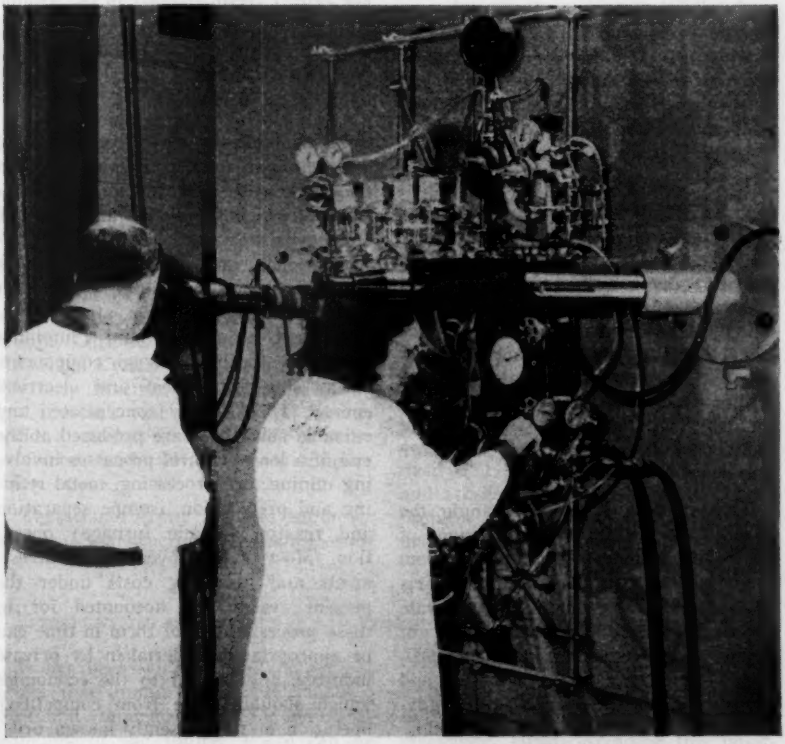
GOVERNMENT OWNERSHIP GREW UNCHECKED BECAUSE OF UNIQUE CIRCUMSTANCES

The birth and early development of usable atomic energy took place under drastically different conditions than those surrounding the beginnings of any other industry of its importance. It broke on the consciousness of the American public all at once, having been the subject of one of the world's best kept secrets. The dual role which it was to play as a source of devastating military weapons and as an economic resource of tremendous importance was grasped by only a few, and by those only in theoretical terms. Its military importance completely overshadowed its nondestructive possibilities in the minds of most.

In terms of financial investment, atomic energy was a giant when the

public first learned about it. Taken in conjunction with the prevailing ideas regarding security control, this fact had major influence on those provisions of the Atomic Eergy Act of 1946 which preserved monopoly of ownership in the government. The adoption of such a national policy, under the circumstances, was hardly surprising; it is both surprising and disheartening that in the intervening five years the accumulation of new understanding has not resulted in a stronger movement to place the ownership of non-military production in private industry.

The lethargic attitude of the general public, and particularly of industrial leaders and of Congress, toward this unfortunate national policy is doubly serious because of what it portends for the eventual state of the atomic energy industry and for other industries which may be influenced by its fate. The lack of public clamor can only be accounted for by a state of unawareness and confusion.



Protection for experimenters is provided by this "hot cell" in which handling of radioactive materials is done by remote control. One of three such cells at Brookhaven, it is lined with stainless steel backed with concrete walls three feet thick. The doors in which the periscopes are set are hydraulically operated, self-sealing slabs of steel one foot thick and weighing eleven tons.

It is not hard to state the reasons for this lack of public awareness; it will be extremely difficult on the other hand to undo or remove them. The most sweeping reasons, referred to many times by David Lilienthal, is the nearly universal attitude that "atomic energy is not my business." It is the business of the military and of a few high civilian officials of the government. This pervading idea has prevented all but the hardiest from making any serious effort to become informed.

As a practical result, few people outside the ranks of the government and its contractors have the degree of knowledge of atomic energy's economic and political problems which citizens normally have about important national issues. This is lamentable but it is, nevertheless, a fact.

Is there still time to change the economic and political orientation of this vastly ramified industry? Are the reasons for government ownership so overwhelming as to make efforts at this late date futile? There is still time and the obstacles while great are not insuperable. But every month of delay and every million dollars of new government investment will make the undertaking more difficult and the outcome more uncertain.

The problem is to disentangle the strictly military application portions of the chain of industrial processes from the whole and to convert all other parts in some equitable manner to private ownership. The actual assembling of atomic weapons using components produced in privately owned plants can and probably should be carried out in government arsenals. Here new developments, improvements, final assembling and testing can be carried on as a related program and under complete government security control. But the production of fissionable components, should be financed and managed by private agencies on a competitive basis.

For this change to have a practical chance of success will require an understanding on the part of industrial, governmental and financial leaders of the economics of the industry and of its future commercial possibilities. Unless it can be seen that there is an opportunity to protect investments and to earn a return, the necessary incentive will not exist. Certain changes in the Atomic Energy Act and in governmental executive policies can make the development possibilities inherent in the industry apparent to investors.

In principle the basic problem is no different here than in older, big industries such as the steel and explosives industries which supply the materials for older type weapons. The technical, organizational and economic problems are different but they can be worked out by a sufficiently positive attack. Fissionable Uranium 235 and Plutonium can be used either as bomb components or as sources of heat and electrical energy. These highly concentrated and valuable substances are produced at the end of a long chain of processes involving mining, ore processing, metal refining and preparation, isotope separation and reactor (atomic furnace) operation. Most of the government's investments and operating costs under the present system are accounted for in these processes. All of them in time can be appropriately undertaken by private industry. In addition to the economies which should derive from competitive operation of the presently known processes there are expected to be many by-product benefits from such unused sources as radioactive wastes or from new discoveries.

PRESENT INDUSTRIAL ACTIVITY IN THE ATOMIC ENERGY FIELD

If atomic energy could have had a normal industrial development, the customary working out of the enterprise system would have resulted during the last five years, first in the wide accumulation by executives of essential business facts and, second, in steps to develop profitable investment. This has taken place only to a limited degree. Private investment in enterprises related at all directly to the atomic energy development has been completely dwarfed by the government's expenditures. These have been at an annual rate of approximately three quarters of a billion dollars. Authorizations since the Korean episode have greatly increased these amounts.

The Atomic Energy Commission's policy of utilizing the skills of industry through contracts has brought a considerable number of firms into atomic energy work as contract managers or as vendors. Four groups or classes of business are involved:

In the first class are the few big operating contractors: General Electric, Union Carbide, du Pont, Monsanto, American Cyanamid, Westinghouse and Western Electric. These operate the government's vast facilities and laboratories at modest financial profit or at no profit but with advantages in gaining know-how.

The second group includes engineering and construction companies, suppliers of "brick and mortar" and thousands of staple and specially made products to the government and its prime contractors. These have enjoyed added volume in their normal lines of business. Operations for this group as for those in the first have involved a minimum of risk-taking and no non-recoverable investment.

In group three is a sizable number of firms which have worked under contract on development of specific machinery, materials and engineering design for the government. While these jobs have been the source of substantial dollar volume for the firms involved and have helped in spreading technical knowledge, the risks and the initiative have been assumed by the government.

The fourth group which is comparatively small is made up of new entrepreneurs who have founded businesses on such things as the sale of instruments or the sale of technical service connected with the use of isotopes. New departments started by already established firms belong in this group. Distinguishing this group from the others is the success of their company managements in finding and seizing opportunities to invest risk-capital in something connected with the atomic energy "industry." How to bring about expansion and acceleration in this direction is the problem which should be faced as a major issue of national policy.

OPPORTUNITIES FOR INVESTMENT BY INDUSTRY AND OBSTACLES TO BE OVERCOME

It has been widely stated in general terms that atomic energy will have a profound influence on the industrial economy of the future. There can be little doubt that the accuracy of this prediction will be borne out. The questions are "when" and "how." Today there are many unknowns in the situation, the largest of which concerns the kind of relationship which will finally be worked out between government and private enterprise. Notwithstanding this unsettled major issue there are even now many more opportunities for private firms to enter some branch of the atomic energy field than is commonly believed.

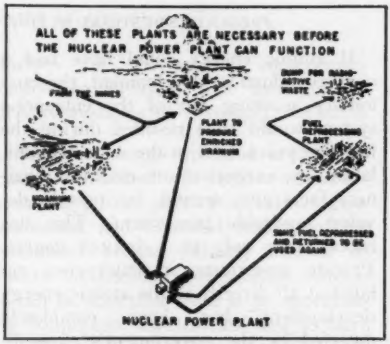
This statement is made in the face of knowledge that businessmen generally believe that profit possibilities in atomic energy are unattractive and that obstacles are too great. These beliefs stem from the prevailing aura of mystery and from lack of information. The absence of a more positive attitude and of more aggressive efforts to enter the field in some manner is undoubtedly attributable to them.

Security restrictions seem to head the list of obstacles in the minds of many. These relate to limited availability of information, delays in clearing personnel, costs of policing operations and of safeguarding fissionable materials. The actual impediments in this situation are not as great as they are often assumed to be, and they can be further reduced by constructive proposals from industry's side.

The hazards of radioactive damage to personnel and property is another deterring factor. Protective measures such as shielding of apparatus, protective control of radioactive wastes and the special training of personnel give rise to unusual costs. But the problems are proving manageable and the monetary costs tend to bear alike on all participants in the field.

The high costs of processing equipment for dealing with radioactive materials, the unavailability of these materials from the government and the shortage of technically trained personnel have been serious obstacles. In addition there has been a tendency to remain aloof from the field on psychological grounds. Possible involvement in F.B.I. investigations tends to dampen enthusiasm for a commercial endeavor when it can be avoided simply by engaging in something else. In general the obstacles to undertaking atomic energy work have actually been less formidable than businessmen have believed them to be.

Too often it is assumed that the generating of electric power is the only


ALL OF THESE PLANTS ARE NECESSARY BEFORE
THE NUCLEAR POWER PLANT CAN FUNCTION
FROM THE MINES
PLANT TO PRODUCE ENRICHED URANIUM
DUMP FOR RADIO-ACTIVE WASTE
FUEL REPROCESSING PLANT
URANIUM PLANT
TAKE FUEL REPROCESSED AND RETURNED TO BE USED AGAIN
NUCLEAR POWER PLANT

A sketch of the complexities of furnishing atomic power for industry, giving some idea of the enormous capital outlay involved.

commercial possibility in the future of atomic energy and that its achievement on an economical basis is several or many years away. Power may or may not prove to be the greatest beneficial product of atomic energy. It is far too early to know what the relative importance of different developments will be. But even the most cautious must see that the sum of the many industrial, medical, research and power uses of this tremendous new energy resource will profoundly affect future industry and life.

Already a business of respectable proportions has been development in the production, refining and distribution of low-energy radio isotopes for industry, hospitals and research institutions. Development and sale of special detecting, measuring and recording instruments for use in radioactive work have progressed rapidly also. A catalogue issued in 1950 listed some eighty-two firms as sources of such instruments.

Exploration of the industrial and research uses of high-energy radioactive materials which exist in great quantities as "by products" of the present production processes, is only just getting started. The potential in this field is not known but may prove to be very large. Its development can get under way in

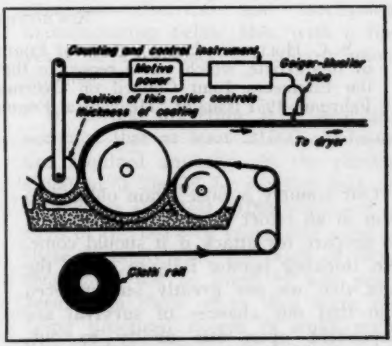
earnest when the government becomes persuaded that it is in the national interest to make materials and technological information available.

Similarly the important field of nuclear reactor operation can have the benefit of competitive cost reduction and industrial engineering efforts when the necessary changes in government policy are made to allow industry to own fissionable materials and reactors.

If the beneficial uses of atomic energy are to have a chance to develop at all rapidly and if this industry is to take its place as part of the free enterprise system, initiative will have to be provided from private sources. The agencies of the government responsible for atomic energy are occupied with operational problems, the accomplishment of which is their principal concern.

It will not be easy to devise plans which will make it profitable for private investors to own and operate the plants now belonging to the government. Nor will it be easy to split up the largest installations so as to avoid undue economic concentration. But these and the many other attendant problems should be taken up responsibly by the business community without further delay.

Solutions can be found. The failure to do so leaves only the alternative of perpetuated, tight government monopoly of this new great energy resource which, properly developed, holds the promise of untold benefits for mankind.



A machine currently in use utilizes radioactive isotopes to measure and control thickness of a microfilm application.

THE DEVELOPING CRITICAL SHORTAGE OF ENGINEERS

ACKNOWLEDGMENT

S. C. HOLLISTER, Dean, College of Engineering, Cornell University, is the author of this article, which was a report to the Engineering Manpower Commission of the Engineers Joint Council on December 18, 1950. It was published in the February 1951 issue of "Mechanical Engineering."

Our country is once again obliged to arm in an effort to preserve peace and to prepare for attack if it should come. No thinking person fails to grasp the fact that we are greatly outnumbered and that our chances of survival are dependent upon our ability to out-design, out-develop, out-produce, and out-perform our enemies. At the same time that we are preparing for possible large-scale war, we must maintain, on a more efficient basis than ever before, our civilian functions. Moreover, we must prepare for eventual attack on our cities and on our industries and for the invasion of our homeland.

World War II brought to public attention the extent of the contribution to the war effort of the scientists and engineers. We could not have fought, much less won, that war without the vast equipment planned, developed, built, and operated by engineers. Not only have the engineers provided the country with the labor-saving devices, the means of transportation, communication, production, and distribution in peacetime, but they have also provided the special versions of all these services for war. The engineers design, develop, produce, and operate the ships, the planes, the guns, missiles, rockets, and ammunition, the tanks and armored vehicles, the radio, radar, sonar, loran, and other devices of communication and detection, the vast transportation facilities needed in warfare. They are clearly indispensable.

But we are short of engineers. As we advance in the war preparations the shortage will be greater. As one contemplates the dependence upon engineers in the task before us the seriousness of a shortage in this part of our manpower becomes evident. It comes about because there has been a falling off in enrollments in engineering colleges, largely due to a widely and erroneously held belief that engineers were in oversupply. The shortage will deepen if engineers are drafted or called up as reservists and used in positions for which engineering training and experience are not mandatory.

The accompanying chart presents the essential factors in the outlook for supply of engineering graduates. The lower curve shows the annual output of the engineering colleges of the country for the academic years 1935-1936 to 1950-1951. It shows that from 1942 through 1945, the country, through its mistaken

manpower policy, all but stopped the operation of engineering education. That is the reason why industry today is not successful in finding critically needed engineers with 5 to 10 years' experience. They were not produced at the rate they should have been.

Beyond 1950 a dotted line shows the estimated number of graduates to be expected if the schools operated without any withdrawals by Selective Service. It represents the most optimistic production possible, at least until 1954, since all classes graduating up to that year are already enrolled in the colleges.

Since the number of graduates depends upon the number entering the colleges, the curve of freshmen is shown. Again is evident the low enrollment of freshmen of the war years, many of whom were taken out of college for military service. Following the war, the veteran benefits program brought the unprecedented number of 91,000 freshmen into the colleges in the fall of 1946. These did not come direct from the high school graduations in that year, but represented an accumulation of high-school graduates who, for several years, had not been permitted because of the draft to proceed with college training. Since 1946 the numbers of freshmen have fallen rapidly until in 1950 there are enrolled about 30,000.

Beyond 1950 there is shown a dotted line indicating estimated freshman enrollment. This estimate was made before the Korean War, and was based on normal peacetime expectancies. It is seen that the actual enrollment for 1950 did not reach the estimate, due largely to widespread rumor that the engineering field was oversupplied.

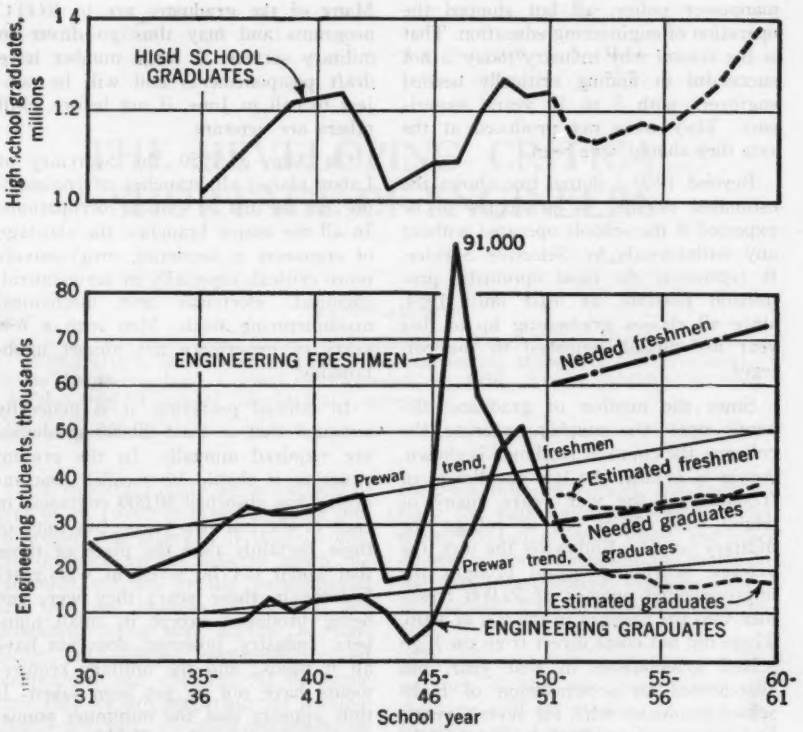
The present situation already shows tightness in all fields and critical shortages in many. Industry has absorbed the last two graduating classes of nearly 50,000 each and is hotly contesting for the 30,000 that will graduate in 1951. Many of the graduates are in ROTC programs and may thus go direct to military service. A large number have draft postponements and will be subject to call in June, if not before. Still others are veterans.

On Aug. 3, 1950, the Secretary of Labor placed all branches of engineering on the list of critical occupations. In all the major branches, the shortage of engineers is becoming progressively more critical, especially in aeronautical, chemical, electrical and mechanical manufacturing fields. Men with a few years of experience are almost unobtainable.

In normal peacetime it is generally accepted that at least 20,000 graduates are required annually. In the present situation it should be recalled that industry has absorbed 50,000 engineers in each of the last two years; but some of these certainly take the place of those that could not be hired in war years because in those years they were not being produced except in small numbers. Industry, however, does not have all it needs; and the military requirements have not as yet been taken. It thus appears that the minimum annual need is not less than 30,000 engineering graduates with use for many more if they should be available.

How do we obtain not less than 30,000 graduates yearly? First of all, to do so requires an annual input of 60,000 freshmen. This is about double the number entering engineering schools this year. How may twice as many high-school graduates be attracted to engineering scools? Are there that many among the high-school graduates who have an interest in engineering and who, at the same time, have the necessary aptitude and abilities?

The numbers of male and female high-school graduates anticipated are shown at the top of the chart. It appears that between 1950 and 1958 there will be a 10 per cent dip in the numbers graduat-



OUTLOOK FOR ENGINEERING GRADUATES

(Actual and estimated enrollment by U. S. Office of Education. Needed freshmen and graduates estimated by S. C. Hollister.)

ing so that the range is between 1,250,000 and 1,130,000. We clearly cannot rely, therefore, upon an increase in output of the high schools to solve the problem.

Except for the war and immediate postwar years, which were abnormal, the numbers entering engineering school since 1935 range from 2.7 to 3.1 per cent of the total high-school graduating classes. On the basis of an average high-school output of 1,200,000 boys and girls, 5 per cent would be required to produce 60,000 engineering freshmen. This is about 10 per cent of the boys graduating from high school. There is some doubt whether the combined factors of aptitude, ability, and interest would be found in so high a percentage. It seems possible that the 60,000 freshmen shown as needed on the chart may be at or above the available number that can be expected to enter engineering. This brings home the clear fact that there is not, and there cannot be developed, an inexhaustible supply of engineering talent, but instead, the supply is sharply limited.

Nevertheless it is essential that all available avenues of information be

used to acquaint the high-school principals, high-school guidance counselors, and the high-school students themselves with the shortage and consequent need of engineers in all categories. It is vital to our security and welfare that such information reach these persons.

What is significant to industry in the picture here set forth? It means that there will not be young replacements for engineering jobs vacated by the calling of reservists or of draftees. There will be only a partial supply of young engineers for new jobs as conversion, retooling, and war production may require. Intercompany pirating will only add to the difficulties of staffing. Robbing the colleges of faculty members for industrial jobs will surely dry up the supply and reduce the quality of training given by the colleges.

It will be essential for industry to re-examine the functions of the engineering personnel and to make certain that each engineer is being used in the most effective manner possible. No engineer should be used in positions others can occupy as well. Supporting personnel, requiring less training, and experience, should be used to spread as far as possible the effectiveness of each engineer.

Even competent supporting personnel will be tight. It will be necessary to begin in-service training of such personnel and to engage women in so far as possible. The training programs of the last war will need to be re-established. None of these steps can be put off. The organization of industries must be streamlined now.

Government agencies, including the armed forces, must take cognizance of the situation. Plans must be developed whereby the engineering manpower needs of the services and of industry may be so co-ordinated that both will be appropriately served from the limited supply. Industry has the job of supporting the military. To assume that the military needs are paramount or exclusive would be a form of national suicide.

A careful planning for utilization of men with engineering training and experience only in categories where such training and experience are mandatory should be undertaken; although such can probably not be wholly achieved, it is in the interest of the total effort that such an objective be approached as nearly as possible. Sound policies relating to deferment and the calling of reservists must be promulgated with the objective of maintaining a balanced utilization.

It is reported in the technical press that the Russians, in their current five-year plan, are scheduled to have produced 150,000 engineering graduates by the end of 1950. This matches our own output for the same period. During the next five years, however, we will drop by one third unless drastic action is taken to increase the supply of engineering freshmen and to continue the training operation in full force.

THE UTILIZATION OF STEEL BALLS FOR LAUNCHING

ACKNOWLEDGMENT

This article describes a Japanese development brought about by the shortage of standard launching lubricants. It appeared in "The Shipbuilder and Marine Engine-Builder," April, 1951.

Since the cessation of hostilities in 1946, acute shortages of materials have been experienced throughout the world and, needless to say, the effect of these shortages has been especially marked in the defeated countries.

In Japan, standard launching lubricants—grease, tallow, paraffin, etc.—have been practically unobtainable. In the early post-war years, the shipbuilders, who were then permitted to build only small vessels, managed to transfer these craft to the water by means of tracks and rollers, and sometimes by heavy-duty cranes.

It soon became evident, however, that, as shipbuilders were allowed to build larger vessels, these expedients would be inadequate, and, after much research, the technical staff of the Yokohama Shipyard and Engine works of the East Japan Heavy Industries, Ltd. (formerly the Mitsubishi Heavy Industries, Ltd.), evolved the steel-ball system, which is now the subject of Japanese Government Patent No. 177,590.

With this system, the launching arrangements are similar to those adopted in standard launching practice, with the exception that, in place of the usual lubricant between the fixed and sliding ways, groups of steel balls are located at specified intervals, the balls being retained in position by special holders. As the vessel moves down the berth the balls rotate in special tracks arranged on the ground and sliding ways. A photograph of a vessel being launched by this method is reproduced in Fig. 1.

Fig. 1.—Launching the "Yama" by Means of Steel Balls.

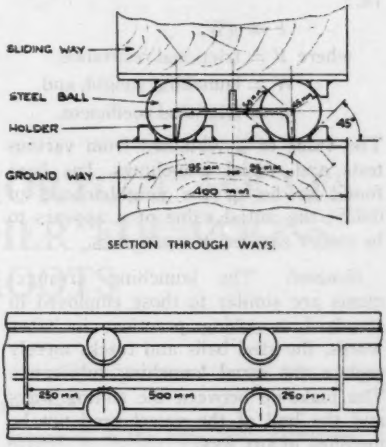
LAUNCHING EQUIPMENT

Steel Balls. The balls are manufactured in high-chrome steel, the finished diameter of each ball being about 3½ in., while its weight is about 6½ lbs. Various tests were applied to check the strength of the balls under compressive loading. The results proved very satisfactory, no change in shape having occurred under a load of about 50 tons.

Holders. Two types of holders are employed, viz., one to accommodate two rows of balls, and one three rows. A drawing of the two-row arrangement is reproduced in Fig. 2, while photographs of the three-row arrangement are reproduced in Figs. 3 and 4. The holders are of inverted channel section, the toes of the flanges resting on the groundway track plate (*q.v.*).

Tracks. The tracks arranged on the groundways and sliding ways consist of mild-steel plate, about 0.7 in. in thickness, to which are welded mild-steel bars (1 in. by 1 in.), chamfered at an angle of 45 deg. on the inside face.

Where the fore poppet accepts the maximum load, i.e., when the stern lifts, the plate thickness is doubled.

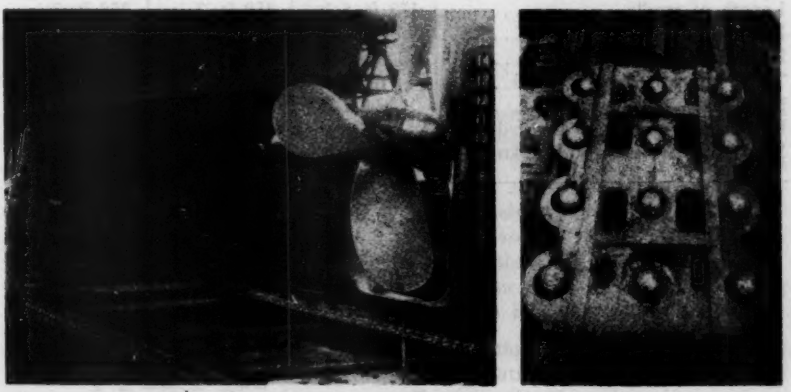


SECTION THROUGH WAYS.

PLAN OF WAYS.

Fig. 2.—Two-row Arrangement of Steel Balls.

Friction of Equipment. The frictional resistance of the equipment may be considered to consist of the rolling friction of the balls and the sliding friction of the holders. Actual data show that the frictional resistance can be re-



Figs. 3 and 4.—Three-row Arrangement of Steel Balls.

garded as being nearly in direct proportion to the launching weight involved, i.e.:

$$F = \mu W,$$

where F = frictional resistance,

W = launching weight, and

μ = frictional coefficient.

The value of μ, obtained from various tests and actual launchings, has been found to be in the neighborhood of 0.025; the initial value of μ appears to be rather smaller in most cases.

General. The launching arrangements are similar to those employed in standard launching practice; in other words, the steel balls and tracks merely replace the usual launching lubricants. The make-up between the sliding ways and the hull of the vessel need not be modified in any way.

Among the advantages claimed for the steel-ball system is that it is independent of temperature variation; therefore, as the possibility of sticking is eliminated, emergency hydraulic-ram equipment to initiate movement is unnecessary.

The ball method of launching has been employed with much success at the Yokohama shipyard since 1947. Launching data for three vessels which have been set afloat by this method are given in the accompanying table. The *Yama*, which was the thirteenth vessel so launched, entered the water on the 26th August, last year, and was waterborne in 41 sec. from the moment of release.

In conclusion, it may be said that the method has proved efficient and economical. The balls, holders and tracks may be used repeatedly, special devices being fitted at the end of the groundways to collect those balls and holders which travel beyond the way-ends.

LAUNCHING DATA OF THREE SHIPS, LAUNCHED FROM THE YOKOHAMA SHIPYARD BY THE STEEL-BALL METHOD

	San Pedro Maru.	*Sakura.*	*Yama.*
Length B.P.	543 ft. 9 in.	442 ft. 11 in.	436 ft. 4 in.
Length of cradle	473 ft. 6 in.	358 ft. 7 in.	358 ft. 7 in.
Declivity of ways	1:20.5	1:20.5	1:20.5
Breadth of ways	3 ft. 0 in.	3 ft. 0 in.	3 ft. 0 in.
Total launching weight, tons	6234	3730	3770
Number of balls used	3402	2616	2616
Initial bearing load on one ball, tons	2.2	1.72	1.74
Maximum launching speed, knots	11.61	10.96	11.40

I—ENGINEER'S SHORTCUT SOLVES CUBIC AND HIGHER EQUATIONS AND ROOTS

ACKNOWLEDGMENT

This reprint consists of three articles appearing in the June 1950, September 1950, and February 1951 issues of "Civil Engineering." The author is DR. D. B. STEINMAN, Consulting Engineer, New York, N. Y. Part I is a condensation of the author's original monograph scheduled to appear in a mathematics quarterly. This condensation elicited many comments from readers, resulting in the preparation of a letter to the editor of "Civil Engineering," reproduced here as Part II. In Part III, the author describes another method for solving the same problems discussed in Parts I and II.

A practical "engineer's shortcut" whereby cubic and higher-degree equations can be speedily solved, and higher-degree roots (cube roots, fifth roots, seventh roots, etc.) of given numbers can be easily extracted, in each case to any desired degree of accuracy has been developed by the writer and is here presented.

SOLUTION OF EQUATIONS

First, the method for solution of equations is explained directly by the use of numerical examples.

Example 1: Cubic Equation

$$x^3 = 5x^2 + 2x + 3$$

5	25	135	740	...
	2	10	54	...
		3	15	...
$r =$ 5	27	148	809	...

$$x = \frac{809}{148} = 5.466$$

The coefficient of x^3 is unity. The other given numerical coefficients (5, 2, 3) are written diagonally across the rows and columns. The rows are written by multiplying the coefficient heading each row by the sequence of values of r, as these values become available. The values of r are written by adding the terms in each respective column.

The values of r are the successive approximations to x, x^2, x^3, x^4, etc. Consequently the successive approxi-

mations to x are given by the converging ratios:

$$x \approx \frac{5}{1}, \frac{27}{5}, \frac{148}{27}, \frac{809}{148}, \ldots$$

or $x \approx 5, 5.4, 5.48, 5.466, \ldots$

Stopping the computation at this point gives the answer correct to four significant figures.

Example 2: Quartic Equation

$$x^4 = 5x^3 + 2$$

5	25	125	625	...
			2	...
$r =$ 5	25	125	627	...

$$x = \frac{627}{125} = 5.016$$

In this example, the coefficients of x^2 and x are zero. The corresponding rows are therefore omitted from the tabulation. The calculation is reduced to the ultimate in simplicity.

Example 3: Quintic Equation

$$x^5 = 10x^4 + x^3 + 2x^2 + 3x + 4$$

10	100	1010	10220	103440	...
	1	10	101	1022	...
		2	20	202	...
			3	30	...
				4	...
$r =$ 10	101	1022	10344	104698	...

$$x = \frac{104{,}698}{10{,}344} = 10.1216$$

Simple arithmetic thus gives the answer correct to six significant figures.

SPECIAL CASE OF UNIT COEFFICIENTS

The application can be further simplified in special cases. Thus, if all the coefficients in the expression for x^n are unity, the tabulation may be omitted, and the values of r may be written directly as a simple sequence in which each term is equal to the sum of the n preceding terms.

Example 4: Quartic Equation

$$x^4 = x^3 + x^2 + x + 1$$

$$r = 1, 1, 2, 4, 8, 15, 29, 56, 108, \ldots$$

$$x = \frac{108}{56} = 1.928$$

In this case, all the coefficients in the expression for x^4 are unity. Accordingly, the values of r are written directly as a simple sequence in which each term is the sum of the *four* preceding terms (since the equation is of the *fourth* degree). In fact, any sequence written by this rule would converge to the same value of x, even if random numbers were used for the initial values of r. The method is self-correcting.

NON-CONVERGING CASE

In all the foregoing examples, the absolute values of x were greater than unity—the necessary condition for convergence. If the absolute value of x is less than unity, substitute $y = 1/x$, by simply reversing the equation.

Example 5: Cubic Equation

$$x^3 = 2x^2 - 5x + 1$$

$$\text{Let } x = \frac{1}{y}$$

$$y^3 = 5y^2 - 2y + 1$$

5	25	115	...
	−2	−10	...
		1	...
$r =$ 5	23	106	...

$$x = \frac{1}{y} = \frac{23}{106} = 0.217$$

SHORTCUT SOLUTION

The transformed equation has the same numerical coefficients as the given equation, but *in reverse order* (with signs changed by transposition). Since the solution requires $1/y$, the *inverse* r-ratios are used to give the reciprocal.

NON-UNIT COEFFICIENT

If the term of highest degree is mx^n instead of x^n, it is not necessary to divide all of the coefficients by m before applying the foregoing procedure. Instead, the summation of each column is divided by m to obtain r.

Example 6: Cubic Equation

$$1.2x^3 = 6x^2 - x + 4$$

	6	30	145	...
$(m = 1.2)$		—1	—5	...
			4	...
$mr =$	6	29	144	...
$r =$	5	24.17	120	

$$x = \frac{120}{24.17} = 4.97$$

In this case, $m = 1.2$. The summations mr are divided by 1.2 to give the values of r, which are used as the multipliers for the succeeding columns.

METHOD OF SPEEDING CONVERGENCE

To expedite convergence, start with $r = 1, x_0, x_0^2, \ldots$, where x_0 is an approximate trial root.

Example 7: Cubic Equation

$$x^3 = 3x^2 + 4x - 1$$

3	48	189	...
4	16	64	...
—1	—1	—4	...

$r =$	1	4	16	63	249 ...

$$x = \frac{249}{63} = 3.95$$

The trial root is $x_0 = 4$; hence 1, 4 16, are used as the first three values of r. By this shortcut, the number of steps is reduced, and the answer (of any desired accuracy) is reached with smaller numbers in the calculation.

TO FIND THE NTH ROOT OF A GIVEN NUMBER

To solve the equation $x^n = N$, transform the equation to supply the missing terms and then solve by the foregoing method of convergents. A convenient transformation is $x = a \pm 1/y$, where a is an approximate root, usually the nearest integer to the required root.

Example 8: Square Root

To solve: $x^2 = 7 (= 3^2 - 2)$

Let: $x = 3 - \frac{1}{y}$; $y^2 = 3y - \frac{1}{2}$

3	9	25.5	...
—0.5		—1.5	...

$r =$	3	8.5	24 ...

$$x = 3 - \frac{8.5}{24} = 2.6458$$

This solution, to five significant figures, is actually speedier than the conventional method of extracting square roots, uses smaller numbers, and gives the answer by slide rule. For roots of higher degree, the classical method is completely eclipsed.

Example 9: Cube Root

To solve: $x^3 = 67 (= 4^3 + 3)$

SHORTCUT SOLUTION

Let $x = 4 + \frac{1}{y}$; $y^3 = 16y^2 + 4y + \frac{1}{3}$

	16	256	...
		4	...
			1/3
$r =$	16	260	...

$$x = 4 + \frac{16}{260} = 4.0615$$

The actual numerical work in this solution, giving the answer (by simple mental arithmetic) to five significant figures, can hardly be matched for brevity.

Example 10: Fifth Root

To solve: $x^5 = 37 (= 2^5 + 5)$

Let $x = 2 + \frac{1}{y}$

$y^5 = 16y^4 + 16y^3 + 8y^2 + 2y + \frac{1}{5}$

	16	256	4352	...
		16	256	...
			8	...
				2 ...
$r =$	16	272	4616	...

$$x = 2 + \frac{272}{4{,}616} = 2.0589$$

The answer is speedily obtained to five significant figures without going beyond the third coefficient.

METHOD IS QUICK AND SIMPLE

For the solution of cubic and higher-degree equations, and for the extraction of cube roots and higher roots of given numbers (without the use of tables), the shortcut method here presented should commend itself to engineers for its speed and simplicity. The actual numerical operation has been reduced to a seemingly irreducible minimum.

II—Roots Solved by Steinman's Algorithm

My shortcut method (termed an "algorithm," meaning a conveniently routinized numerical procedure) yields the root of a higher-degree equation by a sequence of convergents. The convergence may be oscillating, with successive values alternately higher and lower than the limit but with the difference progressively diminishing.

If the sequence does not converge or converges too slowly, transform the equation to augment or reduce the unknown and then apply the algorithm. Convenient transformations include:

1. Substituting $y = 1/x$ (by reversing the equation).
2. Substituting $y = kx$.
3. Substituting $y = x + k$ (using synthetic division).

If the equation has more than one real root, the direct application of the algorithm will yield the numerically highest root; and, with $y = 1/x$, the algorithm will yield the numerically lowest root. After any root x_1 is found, divide the complete expression, $f(x) = 0$, by $(x - x_1)$ to reduce the equation, and repeat the operation to find the remaining roots.

The writer's method has been rigorously derived by writing the solution of a cubic (or higher degree) equation as a repeating continued fraction of novel type and then writing the successive convergents in the form of an algorithm. A highly simplified derivation is as follows: Given an algebraic equation such as $x^3 = ax^2 + bx + c$, then the successive approximations are:

$$
\begin{aligned}
r_1 &= x &&= a && &&= a \\
r_2 &= x^2 &&= ax &&+ b &&= ar_1 + b \\
r_3 &= x^3 &&= ax^2 &&+ bx + c &&= ar_2 + br_1 + c \\
r_4 &= x^4 &&= ax^3 &&+ bx^2 + cx &&= ar_3 + br_2 + cr_1 \\
r_5 &= x^5 &&= ax^4 &&+ bx^3 + cx^2 &&= ar_4 + br_3 + cr_2, \text{ etc.}
\end{aligned}
$$

The implied assumption is that the successive terms in the right-hand side of the original equation are of diminishing importance. Steeply ascending coefficients and reversals of sign may represent gross departure from this assumption so as to prevent convergence. An extreme case is a missing second term presented by $a = 0$. In all such cases, a simple transformation of the original equation eliminates the difficulty. (See examples 8, 9, 10, in the original article.)

The writer's method yields speedier convergence than Newton's Method and greater speed and simplicity than any other known method. The only disadvantage is that a transformation of the original equation is sometimes required. Against this, it should be noted that the conventional solution (by Horner's Method) requires transformation of the equation for each successive significant figure in the answer, in addition to requiring the roots to be found by trial (by synthetic division) at each step.

It is always possible to find or devise problems containing unfavorable relationships so as to place a given method of solution at a seeming disadvantage. Nevertheless, the writer has not encountered any problem in cubic or higher-degree equations that cannot be solved advantageously by his method. The most difficult problems that have been submitted by correspondents are solved by the following examples.

Example D1:

To solve: $d^3 - 14d - 12 = 0$

In this case, the necessary second term is missing. By trial, d is close to 4. Hence, to supply the missing term, let $d = 4 + (1/x)$. This transformation is conveniently given by synthetic division, thus:

4)	1	0	−14	−12
		4	16	8
	1	4	2	−4
		4	32	
	1	8	34	
		4		
	1	12		

Hence, $4x^3 = 34x^2 + 12x + 1$

or $x^3 = 8.5x^2 + 3x + 0.25$

	8.5	72.25	...
		3	...
			0.25
r =	8.5	75.25	...

$$d = 4 + \frac{8.5}{75.25} = 4.113$$

The complete numerical work is given above.

Example D2: To solve:

$$x^4 = -14x^3 + 478x^2 - 686x - 2401$$

In this case the coefficients are steeply ascending, hence a transformation is needed to speed convergence. By trial, x is between 14 and 15. Hence a convenient transformation is $x = 14 + 1/y$. This substitution is carried out by synthetic division as in the preceding example, yielding:

$$4851y^4 = 6510y^3 + 1286y^2 + 70y + 1$$

or

$y^4 = 1.341y^3 + 0.264y^2 + 0.0144y + 0.0002$

1.341	1.800	2.775	
	0.264	0.354	
		0.0144	0.0002
r = 1.341	2.064	3.1434	

$$x = 14 + \frac{2.064}{3.1434} = 14.657$$

No other method will give this answer (to five significant figures) with the same expedition.

Example D3:

To solve: $x^3 = 4.2x^2 - 30.9x + 293$

(This is the problem submitted by Anthony Hoadley, Assoc. M. ASCE, in the August issue, page 49). As in the preceding example (with steeply ascending coefficients and reversals of sign), a transformation is needed to speed convergence. By trial, x is between 6 and 7. Let $x = 6 + (1/y)$. This substitution (by synthetic division, as in Example D1) yields: $42.8y^3 = 88.5y^2 + 13.8y + 1$, or (with slide-rule accuracy)

$y^3 = 2.07y^2 + 0.322y + 0.023$

2.07	4.285	9.536
	0.322	0.665
		0.023
r = 2.07	4.607	10.224

$$x = 6 + \frac{4.607}{10.224} = 6.451$$

Example D4:

To solve: $8x^3 = 9x^2 - 1$

By inspection, $x = 1$. Hence, divide the complete expression by $(x - 1)$, using synthetic division as follows:

1)	8	−9	0	1
		8	−1	−1
	8	−1	−1	0

Hence, $8x^2 = x + 1$. This quadratic equation may be solved either by the conventional method or by the writer's algorithm (substituting $y = x + 1$ or $y = x - 1$), yielding $x = 0.4215$ and $x = -0.2965$. (The sum of the three roots must equal the coefficient 9/8 in the original equation, and their product must equal the coefficient −1/8.

Example D5:

To solve: $4x^3 = 18x^2 - 20x + 3$

The correspondent who submitted this problem found $x = 1.5$ by starting with this value as a trial root in the writer's algorithm, but could not obtain convergence to this root when he started with $x = 1.4$ or 1.6. That is because this equation has a higher root $x = 2.823$, which is given by direct application of the algorithm. To find the third root, substitute $x = (1/y)$ by simply reversing the equation, thus: $3y^3 = 20y^2 - 18y + 4$.

Application of the algorithm to this equation yields $x = 0.177$. When any two of the roots have been found, the third is given by the fact that their sum must equal the coefficient 18/4.

Example D6:

To solve: $x^3 - 100x - 150 = 0$

To obtain coefficients in descending order (for speedier convergence), let $x = 10y$, yielding $y^3 - y - 0.15 = 0$.

To supply the missing second term, let $y = z - 1$, using synthetic division, thus:

−1)	1	0	−1	−0.15
		−1	1	0
	1	−1	0	−0.15
		−1	2	
	1	−2	2	
		−1		
	1	−3		

Hence, $z^3 = 3z^2 - 2z + 0.15$

Applying the algorithm yields $z = 2.07$. Hence $x = 10z - 10 = 10.7$.

To obtain the other roots divide $f(x)$ by $(x - 10.7)$:

10.7)	1	0	—100	—150
		10.7	114	150
	1	10.7	14	0

Hence, $x^2 + 10.7x + 14 = 0$.

Applying the algorithm yields $x = -9.17$. The sum of the three roots must equal the coefficient (zero) of x^2 in the original equation. Hence, the third root is $x = -1.53$. For a check, the product of the three roots equals the absolute term (150) in the original equation.

Example D7:

One correspondent submits this cubic equation arising in his practice in structural design:

$$x^3 = 46.35x^2 - 361.5x - 1400$$

Applying the algorithm yields $x_1 = 34.81$

Divide by $(x - 34.81)$, using synthetic division:

34.81)	1	—46.35	+361.5	+1400
		34.81	—401.7	—1400
	1	—11.54	— 40.2	0

Hence, $x^2 = 11.54x + 40.2$.

Applying the algorithm to this quadratic equation yields $x_2 = 14.344$. The sum of the three roots must equal the first coefficient, 46.35. Hence, $x_3 = -2.804$.

Approximate trial roots, for speeding the convergence, may be found by roughly plotting the equation as $y = f(x)$; or by testing with synthetic division to find change of sign of the remainder. Thus, in the preceding example, synthetic division shows that there must be a root between $x = 34$ and 35, and another between $x = 14$ and 15. Partial application of the algorithm also yields an approximate value of the root, with which a new start may be made for expedited solution.

In my more comprehensive monograph on the subject, the method is extended to include the numerical solution of imaginary roots.

III—Simple Formula Solves All Higher-Degree Equations

Interest in short-cut methods for solving cubic and higher equations and roots is considerable, judging from the many letters received by the writer in connection with his article in the June 1950 issue ("Civil Engineering") presenting his algorithm or shortcut for this purpose. The method was further explained in his letter to the editor in the September issue ("Civil Engineering"). Because of this widespread interest, another method has been devised by the writer for solving the same problems and is here presented.

Given any algebraic equation:

$$mx^n = ax^{n-1} + bx^{n-2} + cx^{n-3} + dx^{n-4} + \ldots \quad (1)$$

Then for any approximate trial root x_0, a close approximation to the exact root is given by

$$x_1 = \frac{a + \frac{2b}{x_0} + \frac{3c}{x_0^2} + \frac{4d}{x_0^3} + \ldots}{m + \frac{b}{x_0^2} + \frac{2c}{x_0^3} + \frac{3d}{x_0^4} + \ldots} \quad (2)$$

By repeating this operation, the solution can be carried to any desired degree of precision. For engineering design purposes, a single application of the formula will usually suffice.

Equation 2 is a universal formula, with no exceptions. It will generally yield speedier convergence than any other known method. It is directly applicable to each real root of an equation. No transformations of the equa-

tions to supply missing terms or to reduce the roots are required. Its simplicity of application is illustrated by the following examples.

Example 1: $x^2 = 3x - 0.5$; $x_0 = 3$.

$$x_1 = \frac{3 - \frac{1}{3}}{1 - \frac{0.5}{9}} = \frac{8(3)}{8.5} = 2.823$$

Example 2: $x^3 = 3x^2 + 4x - 1$; $x_0 = 4$.

$$x_1 = \frac{3 + \frac{8}{4} - \frac{3}{16}}{1 + \frac{4}{16} - \frac{2}{64}} = \frac{77(4)}{78} = 3.949$$

Example 3: $x^4 = 5x^3 + 2$; $x_0 = 5$.

$$x_1 = \frac{5 + \frac{8}{125}}{1 + \frac{6}{625}} = \frac{633(5)}{631} = 5.016$$

Example 4: $x^5 = 10x^4 + x^3 + 2x^2 + 3x + 4$; $x_0 = 10$.

$$x_1 = \frac{10 + \frac{2}{10} + \frac{6}{100} + \frac{12}{1000} + \frac{20}{10000}}{1 + \frac{1}{100} + \frac{4}{1000} + \frac{9}{10000} + \frac{16}{100000}} = \frac{10.274}{1.01506} = 10.1216$$

Example 5 (cube root): $x^3 = 67$; $x_0 = 4$.

$$x_1 = \frac{0 + 201/16}{1 + 134/64} = \frac{201(4)}{198} = 4.061$$

In each of the foregoing numerical examples, the first trial gives the answer correctly to the number of significant figures shown. The following example illustrates the powerful convergence of the method:

Example 6: $2x^3 = 9x^2 + 10x - 12$. (One root, $x = -1.5$) $x_0 = 5$.

$$x_1 = \frac{9 + \frac{20}{5} - \frac{36}{25}}{2 + \frac{10}{25} - \frac{24}{125}} = 5.236$$

$x_0 = 5.236$, $x_1 = 5.2360679775$
(11 significant digits)

The next substitution would yield x to 28 significant figures.

To find the other root, $x_0 = 0.8$

$$x_1 = \frac{9 + \frac{20}{0.8} - \frac{36}{0.64}}{2 + \frac{10}{0.64} - \frac{24}{0.512}} = 0.76$$

$x_0 = 0.76$, $x_1 = 0.7639$
$x_0 = 0.7639$, $x_1 = 0.76393202$
$x_0 = 0.76393202$,
$x_1 = 0.7639320225002103$
(16 significant digits)

The next substitution would yield x to 32 significant figures. The sum of the three roots is $\frac{a}{m} = \frac{9}{2}$, and their product is $\frac{c}{m} = \frac{-12}{2}$.

The foregoing method is directly applicable to negative and fractional roots. In some cases of small fractional roots, it may be desirable (though not necessary) to reverse the equation and to solve for $1/x$.

Example 7: $4x^3 = 18x^2 - 20x + 3$. (One root, $x = 1.5$)

Let $z = \frac{1}{x}$

$3z^3 = 20z^2 - 18z + 4$ (reversed equation)

$z_0 = 6$

$$x_1 = \frac{1}{z_1} =$$

$$\frac{3 - \frac{18}{36} + \frac{8}{216}}{20 - \frac{36}{6} + \frac{12}{36}} = \frac{548}{516(6)} = 0.1770$$

In applying Eq. 2, the numerator and the denominator may be conveniently evaluated by synthetic division.

Example 8: $8x^3 = 20x^2 - 4x + 1$; $x_0 = 2.5$.

2.5)	20	−8	3	0
		50	105	270
	20	42	108	270
2.5)	8	0	−4	2
		20	50	115
	8	20	46	117

$$x_1 = \frac{270}{117} = 2.308$$

The philosophy and the derivation of the method represented by Eq. 2 are illustrated in Figs. 1 and 2. The given Eq. 1 may be written:

$x = f(x)$

$$= \frac{1}{m}\left(a + \frac{b}{x} + \frac{c}{x^2} + \frac{d}{x^3} + \ldots\right) \quad (3)$$

The intersection of $y = f(x)$ and $y = x$ (Figs. 1 and 2) will give the required root. The conventional methods of approach by successive substitution are represented by "staircase iteration" (Fig. 1*a*) or "spiral iteration" (Fig. 1*b*); and if $|f'(x)| > 1$, the iteration becomes a "divergent staircase" or a "divergent spiral" and the corresponding method fails. The writer's formula (Eq. 2) corresponds identically to Figs. 2(a) and 2(b), in which the conventional iteration methods are replaced by "tangent convergence," yielding a positive and more rapid approach to the exact root.

The writer's formula (Eq. 2) also corresponds identically to—and may be derived by—the application of Newton's tangent method to Eq. 3, written in the form,

$y = f(x)$

$$= x - \frac{1}{m}\left(a + \frac{b}{x} + \frac{c}{x^2} + \ldots\right) \quad (4)$$

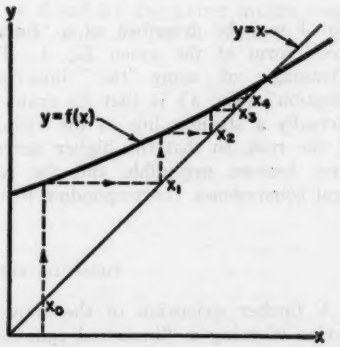


FIG. 1(*a*). Staircase iteration.

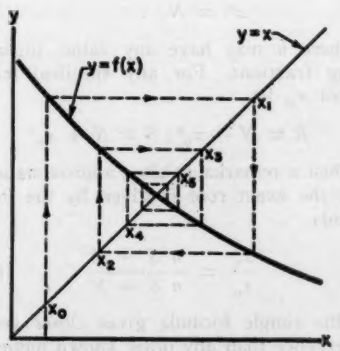


FIG. 1(*b*). Spiral iteration.

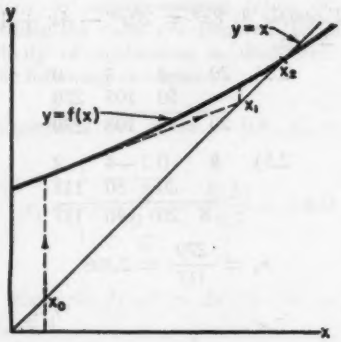


FIG. 2(*a*). Tangent convergence.

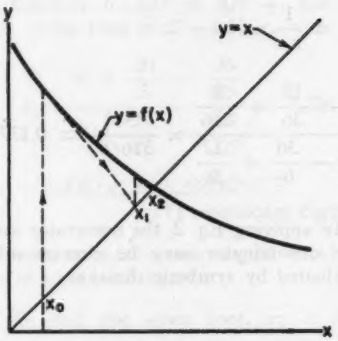


FIG. 2(*b*). Tangent convergence.

Eq. 4 may be described as a "linearized" form of the given Eq. 1. The advantage of using the "linearized equation" (Eq. 4) is that its graph is virtually a straight line in the vicinity of the root, so that the higher derivatives become negligible and the tangent convergence (corresponding to the first derivative) comes very close to the exact root. The application of Newton's tangent method, in the form,

$$x_1 = x_0 - \frac{f(x_0)}{f'(x_0)}$$

to Eq. 4 yields the writer's formula, Eq. 2.

FORMULA FOR ROOTS OF NUMBERS

A further extension of the writer's device of using a "linearized equation" yields a very simple formula for finding any specified roots of given numbers. Given,

$$x^n = N \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where n may have any value, including fractions. For any specified trial root x_0, let

$$R = N - {x_0}^n; \; S = N + {x_0}^n$$

Then a remarkably close approximation to the exact root is given by the formula,

$$\frac{x_1}{x_0} = \frac{n\,S + R}{n\,S - R} \quad . \quad . \quad (6)$$

This simple formula gives closer convergence than any other known method of comparable simplicity and generality. It is derived by applying Newton's tangent method to

$$f(x) = x^{(n+1)/2} - Nx^{-(n-1)/2} = 0 \; . \quad (7)$$

which is the "linearized" form of Eq. 5. For Eq. 7, the second derivative $f''(x)$ is zero at $f(x) = 0$, so that the graph is virtually a straight line in the proximity of the root. Consequently Eq. 6 yields very close convergence, equal to that attainable by including the second derivative in convergence formulas. For engineering design purposes, a single application of Eq. 6 will ordinarily suffice.

Example 9: $x^2 = 152$

$$x_0 = 12) \quad \begin{array}{r} 152 \\ 144 \\ \hline \end{array}$$

$$R = 8, \; S = 296$$

$$\frac{x_1}{12} = \frac{2(296) + 8}{2(296) - 8} = 1 + \frac{2}{73}$$

$$x_1 = 12.3288$$

Example 10: $x^3 = 121$

$$x_0 = 5) \quad \begin{array}{r} 121 \\ 125 \end{array}$$

$$R = -4,\ S = 246$$

$$\frac{x_1}{5} = \frac{3(246) - 4}{3(246) + 4} = 1 - \frac{4}{371}$$

$$x_1 = 4.9461$$

Example 11: $x^7 = 2523$

$$x_0 = 3) \quad \begin{array}{r} 2523 \\ 2187 \end{array}$$

$$R = 336,\ S = 4710$$

$$\frac{x_1}{3} = 1 + \frac{2(336)}{7(4710) - 336} = 1 + \frac{16}{777}$$

$$x_1 = 3.0618 \ (3.0619)$$

Example 12: $x^{4/3} = 85$

$$x_0 = 27) \quad \begin{array}{r} 85 \\ 81 \end{array}$$

$$R = 4,\ S = 166$$

$$\frac{x_1}{27} = 1 + \frac{2(4)}{\frac{4}{3}(166) - 4} = 1 + \frac{6}{163}$$

$$x_1 = 27.994$$

In submitting his formula solutions (Eqs. 2 and 6), the writer invites comparison with any known methods.

TORSIONAL VIBRATION PROBLEMS OF GEARED DIESEL INSTALLATIONS

ACKNOWLEDGMENT

This article appeared in the April, 1951, edition of "The Marine Engineer and Naval Architect." It discusses the torsional vibration problems of some rigidly-coupled geared diesel installations now being built in Europe.

An interesting feature of a series of Werkspoor geared-diesel engine installations, five of which are intended for foreign cargo ships, is the absence of any form of flexible coupling—hydraulic, electric, or purely mechanical—between engine and gearbox, a type of installation which revives memories of the few similar geared plants which Blohm & Voss built with, it was stated, good results more than twenty years ago. The engines of the modern examples are Werkspoor type TMABS 398 four-strokes, developing 1,700 bhp at 275 rpm in eight cylinders 15.35 in. (390 mm) bore and 26.77 in. (680 mm) stroke. They are of the trunk-piston type and are fitted with Brown-Boveri turbo-chargers. The power is transmitted through a Maag type G.85 gearbox which reduces the engine revolutions in the ratio 275/112.6 to drive a four-bladed propeller 13 ft. 1½ in. (4000 mm) in diameter. The teeth of the pinions and wheels are hardened and ground on Maag tooth-profile grinding machines and the general assembly is shown in the accompanying illustration. The crankshaft is coupled rigidly to the pinion-shaft and the whole is located axially within the gearbox, i.e., there is no thrust bearing within the engine itself. The wheel-shaft is coupled rigidly to the propeller shaft and the whole is located axially by the main thrust bearing, separate from the gearbox. Single-helical teeth are used, to permit axial movement between wheel and pinion, i.e., between the driven and driving sections of the system, thus eliminating the need for a sliding coupling between engine and gearbox.

The diagram of torsional vibration stresses has been prepared from data supplied by the engine builders and shows that so far as shaft stresses are concerned it has been possible to obtain a favorable disposition of the significant critical zones without resorting to a special coupling between engine and gearbox. Practical experience has amply confirmed the general theory of torsional vibration which indicates that in an engine aggregate of this type there are usually two significant modes of vibration, namely the fundamental mode with one node in the vibrating system and the first higher mode with two nodes. These are sometimes referred to as torsional vibration of the first and second degrees respectively, designated in the accompanying diagram by the Roman numerals I and II. Experience and theory also indicate that the principal excitations in systems of the type under consideration occur at frequencies which are integral multiples of the fundamental engine firing-impulse frequency for engine excitations, and at integral multiples of the number of propeller blades for propeller excitations.

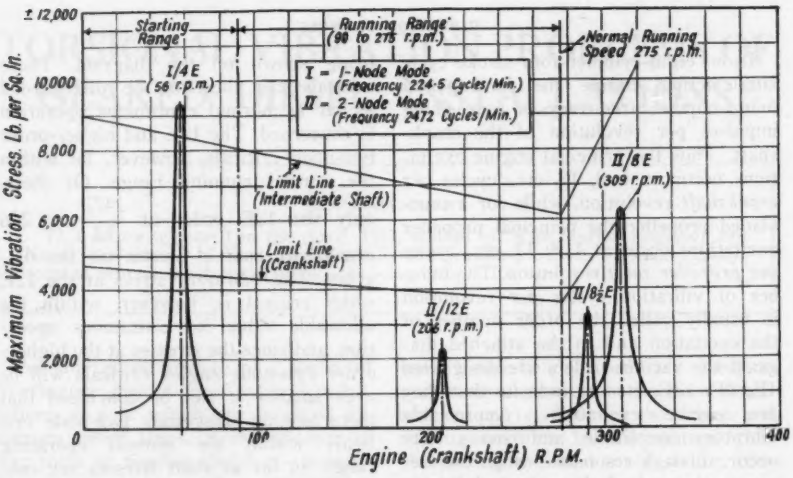
VIBRATION SOURCES

In an eight-cylinder four-stroke cycle single-acting engine the fundamental firing-impulse frequency is four firing impulses per revolution of the crankshaft. Thus the principal engine excitations occur at 4, 8, 12, etc., cycles per *crankshaft* revolution, while for a four-bladed propeller the principal propeller excitations occur at 4, 8, 12, etc., cycles per *propeller shaft* revolution. The number of vibration cycles per revolution is usually called the order number of the excitation and in the attached diagram the various orders are designated 4E, 8E, 12E, etc., to indicate that they are engine excitations. Appreciable vibratory amplitude and stress may occur, through resonance, when the frequency of any of the principal engine or propeller excitations coincides with the frequency of one of the natural modes of vibration of the system. The speeds in rpm at which a resonant response is obtained are usually called critical speeds and are determined by dividing the natural frequencies by the excitation order number. In the installation under consideration the natural frequency of the one-node mode of vibration is 224.5 cycles per minute, i.e., the fourth engine order critical speed corresponding to this mode occurs at $\frac{224.5}{4}$, or 56 crankshaft rpm., and is designated in the attached diagram by the symbol 1/4E. This is, in fact, the only significant engine-excited one-node critical, and since it occurs well below the running range (90 to 275 crankshaft rpm.) it can be disregarded so far as continuous operation is concerned. The 8E and higher engine-order one-node criticals lie even further below the normal running range.

With regard to two-node criticals, the 4E and 8E orders lie above the normal running range, only the 8E order at $\frac{2472}{8} = 309$ crankshaft rpm being shown on the diagram. These criticals can therefore be disregarded so far as normal continuous operation is concerned. The 12E and higher-order two-node criticals, however, lie within the normal running range. Of these, only the 12E order at $\frac{2472}{12} = 206$ crankshaft rpm is shown on the diagram. The vibratory stress at the 12E order critical is, however, within the allowable value for continuous operation, and since the stresses at the higher-order two-node engine criticals will be even smaller it can be concluded that there are no dangerous two-node criticals within the normal operating range, so far as shaft stresses are concerned.

The "Limit lines" shown in the diagram are the limits of torsional vibration stress for continuous operation, proposed by Dr. S. F. Dorey in his presidential address to the Institution of Mechanical Engineers in October, 1950. The lower line refers to stresses in the crankshaft, while the upper line refers to the intermediate shaft. Dr. Dorey also gave values for the permissible vibratory stresses under transient conditions, i.e., during starting or stopping, or during a momentary overspeed condition. These transient stresses are appreciably higher than the values for continuous operation.

In the installations under consideration the one-node vibratory stress is greatest in the intermediate shaft (10.24 in. or 260 mm. diameter) and is small in the crankshaft. The one-node vibratory stresses should therefore be compared with the limit line for the intermediate shaft. On the other hand, the two-node vibratory stress is greatest at No. 6 crankpin (11.81 in. or 300 mm. diameter and 6.1 in. or 155 mm. bore), and is small in the intermediate shaft. The two-node vibratory stresses



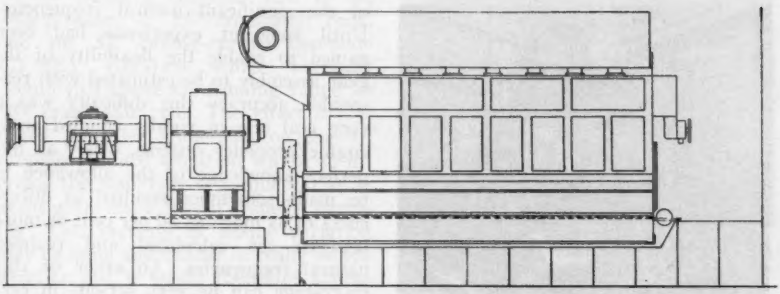
This diagram of the torsional vibration stresses in the engine under discussion also shows the "Limit lines" proposed by Dr. Dorey.

should therefore be compared with the limit line for the crankshaft.

Examination of the diagram shows that the vibratory stresses at all the significant engine-excited criticals, both one- and two-node, are within or very close to the limits proposed by Dorey for continuous operation. Bearing in mind that the higher stresses are for criticals outside the normal running range and that much higher limiting values are permissible for such transient conditions, it is evident that so far as shaft stressing is concerned the disposition of criticals in these installations is favorable.

Incidentally the 8½E critical shown on the diagram is the largest of the so-called "minor" criticals which torsional vibration theory and practice indicate to be a possible source of danger in connection with two-node vibrations in certain cases. Their relative magnitude depends upon the firing order of the engine and the shape of the engine portion of the two-node elastic curve. In the present example the vibratory stresses corresponding to "minor" orders higher than the 8½E will probably be smaller than the value for the 12E principal order.

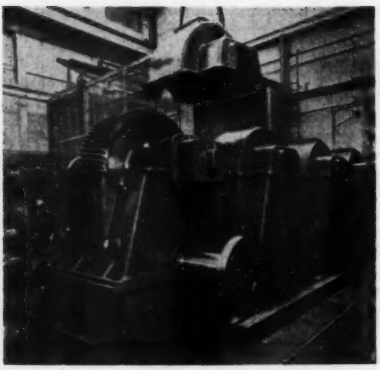
The data supplied by the engine builders contains no reference to vibratory stresses arising from propeller action. As already mentioned the principal propeller excitations are the 4P, 8P, 12P, etc. orders for a four-bladed propeller. In determining the effect of propeller excitations only the one-node mode of vibration need be considered since for two-node vibration the relative amplitude at the propeller is small. The significant critical speeds for propeller excitation are therefore 224/4 = 56, 224/8 = 28, etc., *propeller shaft* rpm, or 137 and 68.5 *crankshaft* rpm. Of these the 4P order at 137 crankshaft rpm occurs within the normal running range, and it would seem prudent to make sure that the vibratory stress in the screw and intermediate shafting due to this propeller excitation is within permissible limits for continuous operation.



Arrangement of engine, gearcase and thrust block as installed on board

So far only the effect of engine and propeller excitations on shaft stressing has been discussed. Equally important is the effect of these excitations on gear tooth strength with respect to both bending stresses and surface loading. From the point of view of gear tooth operation engine-excited vibration is usually much more severe than that arising from propeller action, because even though the shaft stresses caused by engine-excited vibration are within the permissible limit for continuous operation, the vibratory torque is usually in excess of the mean transmitted torque and with a rigidly-coupled system this might result in tooth separation with consequent danger of severe shock loading of the teeth. For this reason a different type of torsional vibration stress diagram from that reproduced here is desirable when assessing the merits of a geared installation, namely, a diagram in which the summation of the peaks and flanks of the principal resonance zones is shown superimposed upon a line representing the mean transmitted torque. Such a diagram shows at once whether there is any severe torque reversal in the running range of speeds. It is mainly with the object of avoiding severe shock loading at the teeth faces due to excessive torque variation under resonant vibration that some form of flexible coupling is generally employed between the engine and the gearbox. This coupling may be of the hydraulic or electro-dynamic types, in which cases the flexibility is very large and the coupling possesses an appreciable degree of damping. In other cases a purely mechanical (resilient) coupling may be used, with or without some inherent damping. Incidentally it is commonly believed that a hydraulic or electrical coupling completely isolates the gearing from the engine so far as transmission of engine-excited vibration is concerned. This belief is fallacious. Both types of coupling possess restoring forces which tend to bring back the driving and driven parts to their original relationship when they are momentarily disturbed by a vibratory impulse.

In hydraulic couplings the restoring or stabilizing action just mentioned results from the effect of centrifugal force on the fluid circulating between the vanes of the driving and driven members and is therefore proportional to the square of the rotational speed (rpm) of the coupling. Thus the effective stiffness of a hydraulic coupling varies quite appreciably with rpm. This is a characteristic of other devices where centrifugal force is the controlling agency, for example, rotating pendulum vibra-



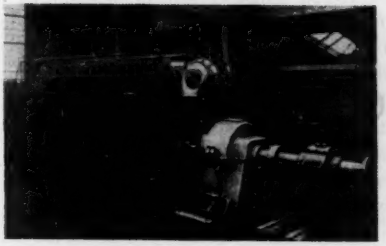
The Maag gear units in course of assembly at Zurich. The output shaft is the upper of the two seen in the nearest gearbox.

tion absorbers. In electro-dynamic couplings the restoring action depends on the electrical characteristics of the assembly and the effective stiffness tends towards a boundary value as the frequency of the applied vibration increases. It is substantially independent of rpm. In the majority of practical examples, however, the effective stiffness of a hydraulic or an electro-dynamic coupling is so small compared with the stiffness of other parts of the system that it can be neglected and the usual assumption that couplings of these types virtually isolate the driven from the driving part of the system is, in general justified by experimental results. The possibility of trouble arising in exceptional cases through neglecting the coupling stiffness should, however, be kept in mind. In particular, if the natural frequency of the driving system happens to be the same as the natural frequency of the driven system, the coupling behaves as though it was rigid.

Another difficulty confronting the designer of a geared installation with a rigid coupling between the engine and the gearbox is that of assessing the true flexibility of the gear assembly, and therefore of determining the true values of the significant natural frequencies. Until sufficient experience had been gained to enable the flexibility of the gear assembly to be estimated with reasonable accuracy this difficulty was a very real one in rigidly coupled aero-engine/propeller systems, where an imperfect knowledge of the allowance to be made sometimes resulted in differences of as much as 30 per cent or more between the calculated and realized natural frequencies. An error of this magnitude can be very serious in certain cases, for example, where a rather severe critical zone lies just outside the running range. Here again, the use of a hydraulic or electro-magnetic coupling having very small stiffness compared with other parts of the system removes the difficulty since the natural frequencies of the driving and driven parts can be estimated separately with the assurance that the calculated and realized values will be in reasonably good agreement. Data on gear flexibility is still very scarce and it would therefore be of considerable value if the builders of the rigidly coupled marine installations under discussion would, in due course, publish the allowances which were made in their particular case together with a comparison of the calculated and realized natural frequencies.

Finally, the question of the effect of adverse weather conditions requires some consideration. It will be recalled that in his paper "Screwshaft Casualties—The Influence of Torsional Vibration and Propeller Immersion," (Trans. I.N.A., Vol. 91, October 1949), Archer directed attention to the possibility of severe overstressing in torsional and bending fatigue during periods of excessive racing occasioned by insufficient propeller immersion, such as might occur in severe weather with the ship in ballast condition. In the geared diesel installations under consideration there appear to be two such danger points namely, the 4P order one-node (¼P) *propeller*-excited vibration, not shown

on the builders' diagram sent to us, which occurs at 137 *crankshaft* rpm., and the 8E *engine*-excited two-node (II/8E) vibration which normally occurs just above the running range, at 309 crankshaft rpm. The effect of reduced propeller immersion on the 4P propeller-excited vibration will be to increase the amplitude of torque variation at the gears due to loss of propeller damping. This increase of torque variation may be quite appreciable in really severe weather and it would therefore appear prudent to avoid operating in the region of the 4P critical zone. In the case of two-node vibration, on the other hand, where propeller damping is not a major damping influence, the principal effect of reduced propeller immersion would be to move the rpm. into the region of the 8E order critical zone and thus expose the gears to considerably increased torque variations. This could be avoided, of course, by operating at sufficiently reduced rpm. in severe weather. It is also possible that the inertia of the revolving parts of the higher-speed diesel engines used in geared installations may be sufficient to prevent excessive racing in severe weather and in this connection it will be recalled that one of the advantages claimed for the use of exhaust turbines in reciprocating steam engine installations is that the inertia of the high-speed turbine rotor prevents racing when the propeller breaks water. The behavior of these rigidly-coupled geared diesel engines in severe weather, especially in the ballast condition, is not yet known, of course, but it is an important consideration on which it is hoped information will be published in due course. Earlier experience with rigidly-coupled geared diesel installations was, we believe, satisfactory and in that belief there is no reason to suppose that these Werkspoor-Maag sets will not be reliable in service. Much knowledge has been accumulated since the earlier German sets were designed and built, better materials are available, and the gears employed are known to be of the highest quality in every respect. The behavior of the installations will therefore be watched with great interest.



Engine and gear unit assembled for text at Amsterdam. The gears have an independent oil cooler.

CRITICAL REVIEW OF GAS TURBINE PROGRESS 1950

ACKNOWLEDGMENT

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The past year was one of consolidation without many outstanding events to record. There were only two new orders for commercial gas turbines placed in Europe, both repeat orders with Brown-Boveri and Co. In the U.S.A., on the other hand, the General Electric Company announces that it has orders for twenty stationary gas turbines, though only two were mobile units. Judging from this solid business, which is being done at apparently very profitable prices, opinion in the U.S.A. must have recovered from its late phase of pessimism and may be about to become as enthusiastic as it was immediately after the war.

This side of the Atlantic most manufacturers and prospective purchasers are apparently prepared to wait and see how the large number of machines at present in construction or early stages of running behave in service before taking on further commitments.

Brown-Boveri and Co., with some eight machines in service, built since the war, is apparently satisfied, after a pause for reconsideration, as to the soundness of its designs and is undertaking further work—the first since 1946. The Beznau machines, after rectification of troubles alluded to last year, have run to schedule without further defect and have now turned out more than 100 million kwh. The 13,000 kw. set started up in February, 1948, has run some 6000 hours, while the 27,000 kw. set has run some 3000 hours since it came into service in January, 1949. As the purpose of the plant is to supplement deficiencies of hydraulic power, the heavy rains in November, which overfilled the rivers, rendered its running unnecessary.

Two other Brown-Boveri machines, whose successful running may be mentioned, are those of 1650 kw. installed in a cement works in Venezuela. Each has done about 5000 hours' running since starting up in September, 1949, and this plant has the distinction of being entirely dependent on gas turbines. A further 5000-kw. unit is now to be added.

A great deal of gas turbine and associated activity is taking place in France, where, counting schemes which originated before the war and also free-piston gas generators, some sixteen projects, are in hand. There is a 5000-kw. two-shaft machine in hand by Alsthom, apparently resembling the American General Electric design of the same size. A 15,000-kw. equipressure boiler, which will employ gas turbines in the air circuit, has been under construction for some time.

Opinion in Great Britain is not so unstable as in the U.S.A. and though

some of the initial enthusiasm has been sobered by the many difficulties encountered in the industrial field, there is little evidence of the attitude that was widespread in the U.S.A. after the period of post-war publicity. A sound understanding of the merits and demerits of different forms of plant, when based on objective considerations, allows excessive enthusiasm to be avoided and its obverse of unreasonable pessimism when unforeseen difficulties are encountered. It is noticeable that troubles with the open cycle seem to have brought about a more tolerant attitude towards the closed cycle, possibly because its difficulties are not so familiar. It may be recorded that mention of the closed cycle no longer engenders visible heat.

NEW PLANT COMING INTO SERVICE

Among the few new machines which came into service in the year, the largest was the 20,000-kw. semi-closed unit built by Sulzer Bros. and installed at Weinfelden, in Switzerland. This did some preliminary running in the Spring, during the period of water power shortage and, as was to be expected in a new design, which had not been shop-tested, a number of troubles were encountered.

The erection of the Escher Wyss 12,500-kw. closed-circuit machine at St. Denis, Paris, has at last been completed and preliminary running is expected to begin immediately. It will be recalled that this machine is similar to that being built by John Brown and Co., Ltd., for the North of Scotland Hydro-Electric Board, for installation at Dundee. There are differences not only in layout but in some of the components, so that the two sets will afford experience on parallel lines. The Dundee machine was due for completion during the year but is evidently somewhat behind schedule. The same apparently applies to the 15,000-kw. open-circuit machines ordered by the British Electricity Authority for Stretford and Dunston stations. A delay of the order of a year seems not unlikely for such machines, bearing in mind their experimental nature and the problems which have to be faced, some of which have been discovered only since the original designs were laid down. Delay is, in fact, the dominant note of the year.

Not counting gas turbine motorcars and motor launches, only two gas turbines were completed in this country during 1950. They are the 1200-hp. British Thomson-Houston set for the tanker *Auris*, and the 3000-hp. experimental set designed by Pametrada and built by the constituent firms for installation in their experimental establishments.

In the U.S.A. a further 3500-kw. machine, built by the General Electric Company, was put into service at Farmingdale. This is a single-shaft unit, employing a plain cycle, and is practically identical with that installed at Arthur Huey station last year and in the Alco locomotive. The Huey plant has now run some 4000 hours on gas without defect. The Farmingdale unit is, however, fired by heavy oil and is claimed to be the first gas turbine of its kind to employ Bunker "C" residual oil. It is intended for peak load service without any heat recovery. The expected efficiency is given as 17 per cent, with an inlet temperature of 1450 deg. Fah., and an air intake temperature of 80 deg. Fah. The price is variously quoted as 132 or 164 dollars per kilowatt.

The same company has announced that it has now obtained orders for twenty units of one or other of its two standard sizes, viz., the above 3500-kw. unit and a 5000-kw. design. This is a two-shaft machine with intercooling and regeneration and shows an efficiency of 28 per cent at 1450 deg. Fah. The sell-

ing price is quoted as 240 dollars per kilowatt. Both machines derive in some degree from aircraft technique in that they are light and small. They thus sacrifice performance to save material and this is reflected in the moderate efficiency of the 3500-kw. unit in relation to its high initial temperature, though not apparently in the price.

Indeed, the prices mentioned above seem to show that the gas turbine in the U.S.A. must have no competitor. They are extraordinarily high in relation to those of other plant whether regarded on per kilowatt of output or per ton of material. They work out at the order of £3000 per ton, whereas steam turbines sell at about £800 per ton and diesel engines at about £500. The prices per kilowatt, which are assumed to cover the gas turbine unit and auxiliaries, are about of the same order as the total cost of large steam stations in the U.S.A., which are much the same as in this country, ranging from 140 dollars to 200 dollars per kilowatt. They include the entire steam power plant. with buildings, civil works, switchgear, cabling, ancillary buildings and site development. On the other hand, the gas turbine price applies to a rather small unit, in which range a steam station would be appreciably higher in first cost and also in cost of attendance. What may seem surprising is that such costly gas turbines can compete either in price or running cost with the diesel engine, since the installations are either in oil-producing regions or at least where oil or gas is available at a price comparable with that of coal. The inference is that the diesel must also be dearer in the U.S.A. than in Europe, despite its wide application in railway traction, or, that the gas turbine has other powerful attractions, such as novelty.

LOCOMOTIVES

The gas turbine locomotive ordered from Brown-Boveri and Co., by the former Great Western Railway arrived in January and has been running on the Western Region lines of British Railways throughout the year though not allocated to any standard schedule. Certain troubles have been encountered, though they do not seem to have interfered greatly with its serviceability. The main points of criticism do not concern the power plant but the heating boiler or the brake compressor. These faults will be rectified in the other gas turbine locomotive with Metropolitan-Vickers equipment, which is approaching completion, and though this set has a considerably higher rating (3500 hp., as against 2500 hp.) the behavior of the two locomotives in service will provide an interesting comparison.

The Alco-G.E. 4500-hp. locomotive, which first took to the rails in November, 1948, had run some 27,800 miles up to November, 1949. It proved dearer to run on diesel fuel than a diesel locomotive, but on Bunker "C" the comparison is stated to be more even. The locomotive was returned to the makers for certain modifications and was later back in service on the Union Pacific.

Three other locomotives with gas turbines are nearing completion in the U.S.A. One is to be powered by a 4000-hp. Westinghouse-Lima Hamilton free-piston engine gas turbine. Another is by the Baldwin Locomotive Company for the Atcheson, Topeka and Santa Fe Railway and has an Elliot gas turbine. The third is an American Locomotive Company's locomotive with Allis Chalmers coal-fired gas turbine, which has been under test for some time. The coal-firing experiments in conjunction with this test are still proceding, a run of some thirty-eight hours having been reported.

VANADIUM ATTACK

Nineteen-fifty will go down in gas turbine history as that of vanadium attack. Previously there had been periods of serious depression over vanadium deposits, particularly in England, but though vanadium attack was suspected by some there was no authentic proof of it until the publication, late in 1949, of a thesis by Paul Amgwerd, of the Zurich Polytechnic. This paper reported researches suggested by Professor Schläpfer on the effects of oil ash containing vanadium in accelerating high temperature corrosion of alloys, particularly those showing a good resistance to scaling in normal atmospheres. It was shown that in the presence of such ash the protection afforded by the oxide layer on stainless steels failed, as the ash apparently acted as a flux at temperatures above 1230 deg. Fah., the melting point of vanadium pentoxide. Further, the action was selective with respect to certain alloy constituents, some materials which were hitherto acceptable for gas turbine blades on grounds of creep resistance and scale resistance showing an alarmingly poor resistance to vanadium attack. Very serious wastage was shown to occur in periods of the order of 100 hours.

Confirmation as to the general validity of the findings was soon forthcoming from other laboratories and, though quantitative agreement is still not reached, the general rule is probably sound that the alloys in which good creep properties are obtained by large additions of cobalt or molybdenum are particularly sensitive; 18/8 stainless iron is fairly resistant and Nimonic 80 is only slightly attacked. Pure chromium is not attacked and hence some hopes are based on heavy chromium-plating of other materials, so as to produce an impervious protecting layer.

Practically all crude oils contain some vanadium, but the percentage varies with the origin. It may be possible to select oils of low vanadium content. An alternative remedy might consist in introducing an inhibitor with the oil and some work has been done along these lines. The trouble is, of course, absent where gas, oil or other distillate fuel is used and those makers who, during the deposit scare, undertook only to burn distillate fuel will be congratulating themselves on their prescience. Attack is also apparently avoided at temperatures below the melting point of vanadium pentoxide—about 1230 deg. Fah.—so lower temperature machines, constituting the greater part of those in service, are expected to be immune from attack if not from deposits, 1200 deg. Fah., which has been widely selected as about the right temperature for open-circuit machines in this country, appears to be uncomfortably near the limit. On the other hand, a number of machines going into service in the U.S.A. will employ temperatures of up to 1450 deg. Fah. and are intended to burn heavy oil. These should, therefore, provide useful evidence in a very short time in actual service as to the immunity or otherwise of the constructional materials employed.

The trouble affects closed-circuit machines also, though in their case it is the air heater tubes which may be expected to waste away. The maximum temperatures selected for the circuit air in the region of 1250 deg. Fah. imply tube temperatures some 50 deg. Fah. higher and therefore in the active range.

A surprising aspect of the discovery is that it should be made only after some thirty years of intensive use of oil fuel in naval and merchant vessels, and also in industry in the U.S.A. and elsewhere. In the light of what is now known, it is clear that certain troubles, such as unexpectedly rapid wasting of uncooled metal in oil-fired boilers, may

be attributed to vanadium attack. The reported burning of the nozzles of the first experimental locomotive gas turbine in the U.S.A. will also be recalled. The machine was run for some 400 hours without ill-effect, but after a further ninety-two hours on heavy oil containing some vanadium some of the nozzles had practically disappeared. This was described, though not explained, as "catastrophic oxidization." The material, a 16 per cent chromium, 25 per cent nickel alloy, with 6 per cent molybdenum, is one which is now known to be very subject to vanadium attack.

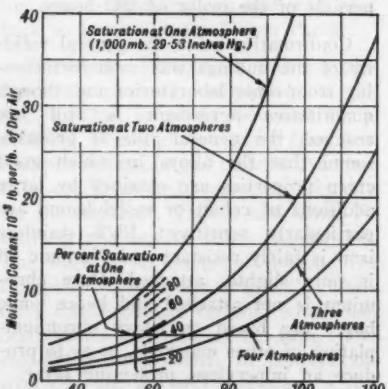
Recently published reports of operation of boilers in the U.S.A., on heavy residual oils, also contain references to severe fouling and corrosion—admitted to be worse than with coal in some cases. Hence the assumption that gas turbines could use heavy oil, because no trouble occurred with fouling in oil-fired boilers, is now proved to have been unjustified.

WATER IN COMPRESSION

Last year mention was made of the interest in wet compression, meaning the proposed introduction of water spray into the air so as to reduce compressor work. This has occurred accidentally in certain compressors and brought to light possible objectionable consequences. Atmospheric air always contains a certain amount of water vapor and may be saturated or unsaturated. The saturation amount depends on temperature and also on pressure being strictly a function of volume. As the volume is reduced by adiabatic compression the temperature rises and there is no inclination for saturation to be exceeded. On the other hand, when the compressed air passes through a recooler, saturation is very likely to be reached and the excess water tends to condense. If provision is not made for draining the water it may be carried forward into the next stage of compression. Its effects there may be, theoretically, beneficial though practically quite the reverse.

In the presence of sulphur oxides inhaled possibly by recirculation of stack emissions and also of CO_2 the water becomes acid and may attack the metals with which it comes into contact. The resulting salts are carried forward in solution and when the water is re-evaporated in the rising temperature air in the next cylinder, the salts are deposited and may foul the blades. Secondly, the water droplets have a braking action on the blades and may damage them.

The issue of how much water is condensed and under what conditions is an interesting problem, which can best be studied by means of a psychrometric diagram, as shown herewith. Such diagrams are familiar in many techniques, such as air conditioning, but are usually drawn for a single barometric pressure. The accompanying diagram is a devel-



MOISTURE CONTENT OF AIR

opment taking into account changing pressure. It is drawn with water content per pound of dry air as ordinate and temperature as abscissa. Processes such as compression and cooling at constant pressure without change of water content are represented by horizontal lines defined by the initial water content. A state of partial saturation can be defined with reasonable accuracy by dividing the ordinate at the required temperature. The diagram shows that on compressing air saturated at 60 deg. Fah. to 4 atmospheres (temperature, say, 370 deg. Fah.) the air becomes much drier. However, on recooling, the saturation line for 4 atmospheres is crossed at 102 deg. Fah., and on further cooling to 80 deg. Fah. an amount of 5.6 by 10^{-3} lb. of water per lb. of air tends to condense out. In a 15,000-kw. gas turbine aspirating 250 lb. per second of air this represents 1.4 lb. per second or 8.4 gallons per minute of water to be drained off. For a more usual case of, say, 80 per cent saturated air at 60 deg. Fah., the rate of condensation will be about 5 gallons per minute or, for 40 deg. Fah. saturated air recooled back to 60 deg. Fah., the yield is about 4 gallons per minute.

That water would accumulate in such quantity was not generally foreseen and the trouble when first encountered was not immediately recognized. It occurs in open-circuit machines particularly, because of the large quantity of air drawn in, and in lesser degree in the semi-closed and closed-circuit types. In the last, only the leakage air has to be replaced so the water quantity is of much smaller order.

FUNDAMENTAL RESEARCHES ON CORROSION

ACKNOWLEDGMENT

The author of this article, which is condensed from a lecture delivered in London on March 28, 1951, under the auspices of the British Iron and Steel Research Association, is DR. U. R. EVANS. It was published in the April 13, 1951, edition of "Engineering."

Corrosion is the cause of much expense and many breakdowns—probably including cases where it is not mentioned in the diagnosis of failure; some fractures which are purely mechanical in their final stages might not have occurred, had there not been pitting or trenching due to corrosion in the early stages, with consequent stress intensification. Empirical remedies are dangerous, since many anti-corrosive methods, beneficial if applied rightly, can intensify attack if applied wrongly; the causes of the trouble must be understood.

The causes have been under study at Cambridge and elsewhere for many years. About 1920, the subject appeared to be a jumble of unintelligible and contradictory facts; today, the facts are seen to obey simple laws, capable of rational explanation. There is a common misapprehension that rust is a result of the direct combination of iron and oxygen. In fact, the phenomenon is much more complex, and some of the intermediate products of corrosion are of considerable importance to the engineer. An example is the sodium hydroxide, caustic soda, that forms during corrosion in salt water, and can attack constituents of some paints. Direct combination does occur, however, in a form well known to engineers.

The direct union of iron and oxygen leads to the scale often found on steel after heating in air. This direct oxidation is only important at high temperatures, since, except where the scale cracks, its thickening requires the passage of iron outwards through the scale to meet the oxygen of the air; such movement in the solid state is slow at low temperatures. If an iron strip is strongly heated at one end, the scale, thick at that hot end, becomes thinner as the colder end is approached. Where the thickness is comparable to the wavelength of light, interference colors are produced; at the cold end, the metal appears unchanged. The slow formation of invisible oxide in dry air at ordinary temperatures has been studied at Cambridge, but possesses no engineering importance.

In general, limited supplies of oxygen favor corrosion by an electrochemical mechanism, whereas ample supplies,

distributed over the whole surface, may suppress attack. Pure iron or zinc "whirled" in pure water containing oxygen remains unattacked, because a continuous protective film is built up. If the whirling ceases, visible attack occurs; if the water is free from oxygen, there is no appreciable attack. On steel or zinc, under engineering conditions, oxygen must generally be regarded as a stimulator of corrosion. On stainless steel, a scanty supply of oxygen suffices to maintain the oxide skin, which protects against many liquids that attack ordinary steel; protection may suddenly fail if the oxygen is shut off completely at certain points. Thus, in choosing materials for industrial plant, the engineer should remember that stainless steel may behave badly under exactly those conditions where many other resistant materials (e.g., Monel metal) behave best.

The fact that small electric currents are flowing over the surface of corroding iron or zinc has been established at Cambridge by several independent methods, which have shown that the currents are sufficiently strong to account for the whole of the corrosion produced. The electrochemical mechanism of corrosion is fact, not theory. It is also clear why electrochemical attack is often so dangerous, whereas direct oxidation, if applied to the whole surface, prevents attack. If the water contains a salt (e.g., sodium chloride) the products formed, both at the anodic (attacked) part and the cathodic (unattacked) part, are freely soluble; a solid substance is formed where they meet and interact, but it will not produce a protective skin and attack will continue. In the case of iron, the two primary products are iron chloride and caustic soda; the secondary product is a mixture of hydrated oxides known as rust. It should be added that rust can be formed in more than one way.

If the salt is one which will produce a sparingly soluble body at either the anodic or cathodic product, even electrochemical corrosion will be very slow. Thus there are two sets of substances which prevent attack, namely, the anodic inhibitors (sodium phosphate, carbonate, silicate, etc.) and the cathodic inhibitors (calcium bicarbonate). If a corrosive salt (sodium chloride) exists in the water, the amount of anodic inhibitor needed is increased, and if the amount added is insufficient, the corroded area will be reduced more than the total destruction of metal; thus the attack is intensified by the ill-judged attempt to "treat" the water. Hence the need for caution.

Important corrosion currents flow when two dissimilar metals are placed in contact in a liquid. The current strength increases if oxygen is supplied to the cathodic metal. If the area of the cathodic metal is large—a factor which will increase the current flowing—and that of the anodic metal is small, the intensity of attack (corrosion per unit area) is great. The cathodic substance need not be a metal; the oxide-scale present on previously heated steel is cathodic towards the steel exposed at tiny cracks in the scale. This combination of large cathode (scale) and small anode (steel) leads to severe pitting in water-pipes or on locomotive boiler tubes. Removal of mill-scale from tubes or plates exposed to waters usually renders attack less intense, though the total destruction of metal is not always diminished.

Similarly, if an attempt is made to protect steel against water or a solution with a layer of copper or nickel (cathodic metals), corrosion is likely to occur if there is a discontinuity in the coating; if the liquid is a highly conducting one, the attack may be locally more intense than if the steel had been left unexposed. This intensification is not so common as is often supposed; it is rarer for nickel coats than copper coats. A nickel coat containing

pores is usually better than no coat at all, but it is best of all to avoid pores in the coat. The frequency of pores diminishes—other things being equal—as thickness increases, but different methods of plating give different pore frequencies even at the same thickness. Work at Cambridge has shown that the surface contour of the basis metal affects porosity. If a scratch-line is traced on steel before nickel-plating, a line of pores is found on the scratch-line; this is due to the depression and not to internal stresses, since annealing before plating hardly affects the frequency of pores. This shows the fallacy of the argument that, if steel is to be plated or enameled, the quality of the steel does not matter.

There is another reason for rejecting this argument. In mildly corrosive environments, the probability that a very small bare point will escape attack is quite appreciable, but the chance of escaping it is greatly diminished if there is much sulphur in the metal, as is shown by statistical work at Cambridge. From the point of view of corrosion, sulphur is undesirable; low-sulphur steel carrying a porous plating is likely to be better than high-sulphur steel with equally porous plating. Once corrosion has started, the effect of sulphur may be either good or bad, according to circumstances; usually it accelerates the anodic reaction and retards the cathodic reaction.

It is often better to coat steel with anodic metal, like zinc, or aluminum. In such cases, after initial irregularities, there is protection to the steel exposed at pores. The zinc or aluminum is attacked, and, at one time, it was thought that protection was purely "sacrificial"; indeed, the wisdom of protecting a cheap material (steel) by the sacrifice of a dear one (zinc or aluminum) has been questioned. In many environments, however, the protection is only sacrificed in the opening stages; soon the cathodic reaction begins to deposit a protective layer (calcium carbonate, zinc or aluminum hydroxide) on the steel and the current begins to slow down, and with it the attack on the coating. A promising method of protecting structural steelwork is to grit-blast the surface, removing scale and rust, and leaving a rough contour suited for interlocking, and, finally, to coat with, say, iron-oxide oil paint. In long-period outdoor exposure tests, organized from Cambridge in several types of atmosphere, this mode of protection gave good results. Even after seven years on a roof in London, the aluminum coating, examined by sectioning, appeared unchanged; the steel was entirely protected. There was no rust, but plenty of soot.

Protection of steel by painting is commonly believed to depend on the exclusion of water and/or oxygen from the metal, but most paint coats are pervious to water and oxygen, and calculation shows that, if the only function of paint was to exclude water and oxygen, it would still permit rapid rusting. However, corrosion, being an electrochemical process, involves movement of ions, and the Cambridge work shows this to take place only very slowly through paint-films; hence the corrosion is slowed down. If, furthermore, inhibitive substances (allied to the inhibitors used in water treatment but less soluble) are incorporated in the innermost coat, corrosion can practically be stopped, provided that there are two or more robust outer coats, chosen to prevent mechanical damage and chemical alteration to the inner coat. Three-coat paint combinations based on this principle have given complete protection during seven years outdoor exposure in London. Even two-coat systems, properly chosen, give fairly good protection if all scale and rust is removed, so as to bring the inhibitive substance into direct contact

with the steel. Comparison of the descaled and scale-bearing specimens in the tests organized from Cambridge show the advantage of proper cleaning of the surface.

Red lead has long been used as an inhibitor. The old-fashioned red lead, which contains the lower oxide PbO as well as Pb_3O_4, gives better protection than non-setting red lead, though it must be applied to the steel soon after mixing with oil; this has often been stated by practical engineers and the Cambridge tests show it to be correct. The addition of PbO to Pb_3O_4 has greatly improved protection. It has also been found that the true inhibitor is not Pb_3O_4 or even PbO, but some substance formed by interaction with the oil. Efforts are being made at Cambridge to identify the "true inhibitor" and if this can be added directly to the paints, without "passenger" substances, economy in lead may reasonably be hoped for.

New series of paints developed at Cambridge are richly pigmented with metallic zinc, either in an organic vehicle (polystyrene) or in an inorganic vehicle (cementiferous paints). The latter paints can be applied to a wet steel surface, and the brush and pot can be washed out with water—a feature which has won appreciation from those who perform the painting. Zinc is in short supply. There are threats of restrictions for the use both of zinc and lead in paint, but the cost of paint coat is so small compared with the cost of labor, etc., in application and the value of the structure or article protected, that even an expensive paint is worth employing if it will increase the period between repaintings.

The corrosion of painted steel follows the same course as that of unpainted steel, but proceeds more slowly. In sea-water, caustic soda is formed at the cathodic regions, and this can destroy some constituents of paint. To obtain good protection against salt water, a paint with an alkali-resistant vehicle must be chosen. This was demonstrated in the Cambridge Laboratory in 1929, and is now being confirmed by raft-tests organized by the British Iron and Steel Research Association. The time lag will be noted. Will it be another 20 years before engineers start to reap benefit from the results?

Fatigue cracking is well known to engineers, but, in the absence of corrosion, it can be easily prevented. There is a stress-range, known as the fatigue limit, below which fatigue failure should never occur. However, in a corrosive environment there is no fatigue limit; any stress, however small, will ultimately cause cracking. This should be borne in mind when selecting an alloy steel for some stressed member, with the intention of reducing cross-sections; alloy steels have often definitely higher tensile strengths and higher fatigue strengths than carbon steels but their corrosion-fatigue properties are often only slightly superior to those of carbon steels. An important exception is the stainless steel group.

Corrosion-fatigue cracking is produced by alternating stress and occurs on almost any material if there is no protection against a corrosive environment. There is another trouble, stress-corrosion cracking, which is produced by steady tensile stress, but which, fortunately, occurs only on some materials after certain heat-treatments; these have generally been established and can usually be avoided. It is commonly stated that corrosion fatigue is recognized by transgranular cracks, and stress corrosion by intergranular cracks, but that is an over-simplification. On magnesium-base aircraft alloys, stress corrosion causes transgranular cracks, while, as shown in some recent Cambridge studies, cor-

rosion-fatigue cracks—although mainly transgranular—may follow the grain boundaries for short distances.

Another Cambridge research has shown the importance of the protection being continuous in time; a short exposure to corrosive conditions during alternating stress, followed by alternating stress without corrosion, may, under some conditions, give a shorter life than if the corrosive substances are allowed to reach the test-piece throughout the whole period of its subjection to corrosive conditions.

Peening (bombardment with small hard particles, leaving a compressed surface layer) is known to give resistance to pure fatigue. A Cambridge study, on high-carbon steel, showed that it improved resistance under corrosion-fatigue conditions against certain liquids, but not all. The benefit was found to vary greatly with the pressure used and the size of the shot; thus careful specification appears necessary. In no cases was the life prolonged indefinitely, and protective coatings may be required on the peened surface. Unfortunately, the contour of the peened surface is unfavorable for the adhesion of certain types of coating. It was found, however, in the Cambridge work, that, if the pressure and shot-size had been chosen to give a thick compressed layer, considerable roughening (by bombardment with angular grit) could be tolerated without loss of the valuable properties left by peening.

A number of practical points emerged from the work discussed. It was important that engineers should understand the causes of corrosion, so that failures really due to this cause were not attributed to mechanical breakdown, as might easily occur where attack was localized.

They should understand also the various effects of oxygen on corrosive action in various conditions, which might sometimes be beneficial and sometimes the reverse, and the electrochemical nature of corrosion, realizing that the intermediate products might have a deleterious effect, e.g., attack on certain paints. Alkali-resistant paints should be used as protection against salt water.

In connection with electrochemical corrosion, they should understand the effect of big cathodes and small anodes and the consequent importance of removing scale. Close attention should be paid to the nature and quality of the steel used, even when it was to be protected. Finally, they should bear in mind the possibility of protecting structures after erection by grit blasting, aluminum spraying, followed by painting with a suitable paint; and of using shot peening, followed by grit blasting and a protective coating, to combat corrosion fatigue.

Experiment seemed to promise that zinc-rich polystyrene paints might be useful where scale was still partly in place, though claims that had been made for performances on thoroughly rusty surfaces were somewhat dubious. Results obtained from the use of aluminum paints as primers, compared with zinc-rich paints, had been somewhat contradictory, and a difficulty arose from the lack of an effective metal-to-metal contact. Work was in hand on this subject. It was to be hoped that restrictions would not be imposed on the use of zinc for protective purposes. The value of the material saved by protection against corrosion was so very much greater than the value of the material expended for this purpose that it would be a false economy were such restrictions to be imposed.

NAVAL GEARING: WAR EXPERIENCE AND PRESENT DEVELOPMENT

ACKNOWLEDGMENT

This article deals principally with the failures in gearing in Royal Navy ships prior to and during World War II, and discusses present developments in the field. It was written by CDR. (E) J. H. JOUGHIN, R.N., and presented at a meeting of the Institution of Mechanical Engineers in London on December 15, 1950. It was published in the January 5 and January 12, 1951, issues of "Engineering."

All but a few of the high-powered naval ships in service during the 1939-45 war were fitted with turbines driving through gearing. The successful use of gearing in the Royal Navy at the end of the 1914-18 war, and subsequently in merchant ships, gave little reason to expect that it would not be absolutely reliable, but two years of war failures showed that where standards of accuracy fell below the average for the country, no positive margin of safety remained. A thorough examination, to which many persons have contributed, has revealed the main causes of these failures. Apart from remedying them, it has been possible to make some immediate advances in design and production standards, to employ stronger materials, and to undertake development work in the expectation that it will provide worth-while advances.

Before the war, the troubles which could occur, and their sources, had been described, but no quantitative measurement of the periodic errors or helical angles had been made. A few firms had given their machines some temperature control, and a proportion of pinions were recut to improve their meshing, but the general practice involved extensive hand work by filing the pinions to bed-in the teeth, followed by the "Parsons" type of lapping. The National Physical Laboratory, however, was developing measuring instruments. Tests on wear and pitting led to an increase of design loading with the Vickers-Bostock-Bramley (V.B.B.) and Parsons "All-Addendum" (A.-A.) tooth forms. These forms gave increased pinion stiffness, allowing the center bearing to be omitted.

When tooth failures occurred in the merchant ship *Strathmore,* her builders drew attention to the dangers to which naval ships were exposed from the high deflection of pinions. Except for *Ajax* and *Griffin,* naval ships differed from *Strathmore* in that they had semi-circular fillets at the roots of the teeth; for the new construction program, improved roots and a standard value for deflection were relied upon. The teeth of ten pinons of ships of the general type of *Glorious* and *Delhi,* with three-bearing pinions, failed by root fractures. All the failures were at the

pinion ends. When a spare pinion was fitted to *York* in 1939, the error in parallelism of the bores of the gear-case was found to be some 0.004 in., but one pinion bearing had to be adjusted to set the pinion axis 0.016 in. out of parallel to ensure even meshing. Fig. 1 shows teeth failure experienced in *Delhi*. The general fatigue failure of the port-inner low-pressure pinion was revealed by turning off the teeth at the aft end of the after helix.

FIG. 1. FATIGUE FAILURE OF PINION.

In 1935, a proposal to adopt larger-pitch teeth for aircraft carriers was abandoned because of the opinion that the limit of load was not determined by the strength of the teeth but solely by the ability of their surfaces to resist deformation or distress. In 1937, a similar proposal for *Illustrious* was abandoned because of the doubts raised by the occurrence in some Italian cruisers of scuffing of gears having 1-in. pitch teeth.

While great distances were sometimes covered during the war, the larger ships operated at high powers for only a small part of the time, whereas some smaller ships spent quite a large proportion of their service at high powers. Merchant ships run continuously at a high percentage of their full power, and if their gear teeth are overloaded, failure will probably occur in the first few months of service, as with *Strathmore*. With naval ships,

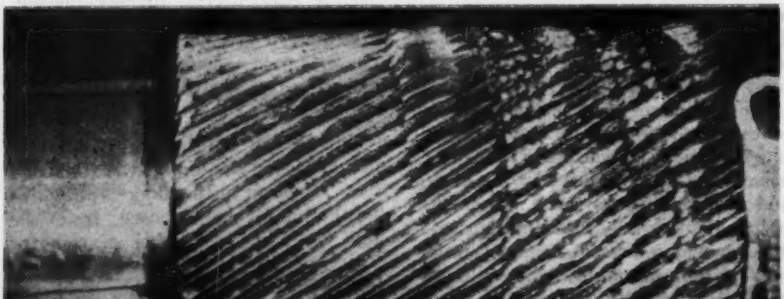


FIG. 2. PORT INNER HIGH-PRESSURE PINION OF H.M.S. "SHEFFIELD."

gear failure may occur at a vital moment after the gears have given years of satisfactory service. This happened with *Sheffield*: when her gears were wrecked she was engaging the *Scharnhorst* in the action in which the latter ship was sunk. Fig. 2 shows the condition of the port-inner high-pressure pinion of *Sheffield* after failure.

During the 1939-45 war the three main sources of gear trouble were: vibration, scuffing, and breaking at the roots. Pitting and water in the lubricating oil were minor sources of trouble. The screaming note of certain non-creep gearing, long since identified with worm error, was experienced in *Scorpion* and *Kipling*. In *Scorpion* the maximum overall noise-level was 113 to 116 British Standard phons. Owing to the abnormally high intensity of the chief component, its level was outside the range of the analyzer used. The exceptionally high noise-level was almost entirely due to the note corresponding to the number of teeth in the hobbing-machine wormwheel. Harmonics of the main note, and others, were also present but were of secondary importance. The banded marking corresponding to the main note was very clear. No quantitative measurements of noise were made in *Kipling*, but both wheels showed the banded marking characteristic of non-creep machines with periodic drive errors. Each tooth of the hobbing-machine wormwheel had impressed a "band" on the wheels, giving rise to a frequency, at full power, of approximately 1220 cycles per second. By recutting the gears on a creep machine the noise was reduced to a satisfactory level.

It had been hoped that with the gear sets of *Scorpion* and *Kipling*, measurement of the motion of the worm-table drive had ensured the reduction of overall errors to an acceptable value. This hope was not fulfilled, however, owing to errors of concentricity in the bevel drive to the worm. Turbine-blade failures in aircraft escort-carriers revealed the magnitude of the vibrating forces imposed by such errors on components coupled to the gears. Measurements of the axial and torsional movements of the pinions under similar conditions were later made, using Sperry and other equipment. The interpretation of these measurements requires care, and it is desirable to examine the manner in which typical errors can produce periodic exciting forces.

When several teeth are in contact at once, the errors on the teeth arising from errors in the lead-screw thrust, the creep ring, or worm errors in creep machines, are not generally in phase. These errors occur in either parallel or spiral bands around the circumference of the gears. The bands of errors are short in pitch and so are only slightly relieved by the local bending of the teeth, and, because they are out of phase, the resulting periodic force has a frequency of (number of teeth in contact) $\times$ (bands engaged per second by any one tooth). The errors produced by the worm error of non-creep cut gears are in phase on all teeth, and the lines of the crests and hollows are roughly parallel to the line of contact of pinion and wheel. In the case where the combined radius of curvature of pinion and wheel surfaces is less than that of the valleys of the error, when the inertia of the system is infinite, the additional load imposed on the teeth is given by the product (amplitude of error) $\times$ (tooth stiffness). Reduction in inertia results in reduction in load. The value of the periodic force will be a small fraction of that given by this product.

These considerations can be applied to the case of *Cavalier*, which had a gear similar to that of *Kipling*. The amplitude of undulation was reported to be 0.000342 in. Assuming a tooth

stiffness of 3×10^{-4} in. per 1000 lb. per inch of face width, and that the non-creep errors were in phase on the two helices, the additional load per inch caused by the error is $(34.2 \times 10^{-5})/(3 \times 10^{-7}) = 1140$ lb. per inch, which is one-quarter of the loading at the ends of the helices. The periodic exciting torque imposed on the pinion and turbine is then approximately one-quarter of the full-power torque. These additional periodic loads on the gear teeth are reduced by any reduction in stiffness of the coupling between the pinions and the driving member. The gears of *Creole,* a sister ship to *Cavalier,* were cut on a non-creep machine having 700 teeth in the master wormwheel, and they had an amplitude of worm undulation of less than 0.0002 in. after hobbing, and still less when shaved. On trials they showed the merest trace of vibration and noise from this error. When the combined radius of curvature of pinion and wheel surfaces is greater than that of the "valleys" of the error, the disturbing force is no longer simple harmonic, and the energy is spread over a multiplicity of frequencies.

During the year before the 1939-45 war, scuffing occurred in five ships. Four of these, *Egret, Pelican, Speedy,* and *Sphinx,* had gears cut on non-creep machines, but the fifth, *Shearwater,* had gears cut on a creep machine. In *Shearwater* there was evidence of misalignment, so that, at the beginning of the war, the assumption that failures would be confined to gears of small ships cut on non-creep machines was justifiable. Fig 3. shows the starboard high-pressure pinion of *Shearwater* after scuffing had taken place. The appearance is typical of the defect.

In this group the significant features are: (1) Failure of the pinion surface occurred on the pattern of high spots imposed by errors in the hobbing machine. On the wheel, damage was not limited to these spots. The phenomenon was repeated in all later cases except that of *Quorn.* (2) Lapping out high spots on the pinion, as in the case of *Pelican,* reduced scuffing but did not guarantee its elimination. (3) In *Speedy* and *Sphinx* there was evidence of concentration of load at one end. (4) Hand work was sufficient to make the gears run satisfactorily.

This type of failure is essentially one of lubrication. Heat generation at local high points and the high local-loading cause breakdown of the oil film and produce metal-to-metal contact with seizure of the two surfaces. Failure starts on the high spots on the pinion, because they are in more frequent contact.

FIG. 3. TYPICAL SCUFFING.

In May, 1940, gear failure occurred in *Fernie,* the first of several failures in the Hunt-class destroyers over a short period of time. An undulation recorder was completed in time to be used in the subsequent examination. The effect of the undulations, which arose from variations in temperature of the gear-hobbing machine while cutting, was a significant feature. These undulations are frequently referred to as "daily bands." As several firms were cutting gears, a review was made of their methods, and of the undulations given by their machines. This

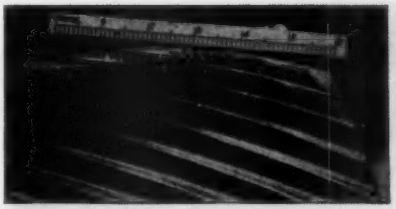


FIG. 4. SCUFFING DUE TO LOCAL INTENSITY OF LOADING.

review showed that gears cut by the firms using the best standards of meshing and temperature control had the lowest incidence of scuffing. In addition to the common errors revealed at that time, a high proportion of recuts were found necessary to match helix angles, but the two firms with the best temperature-control and meshing arrangements did no recutting.

Gear scuffing now occurred in *Quorn*, showing that mere lapping of pinion teeth did not remove long waves. Fig. 4 shows the scuffing due to local intensity of loading on the port high-pressure pinion. The combination of daily bands, misalignment, and deflection of the pinion under load, led to a local intensity of loading with consequent scuffing. The method of ensuring that no warp of the gearcase occurred in the ship was suspected, particularly as *Shearwater* and *Quorn* were fitted out at the same yard. In March, April, and May, 1941, gear scuffing occurred in *Sutlej*, *Jumna*, and *Widgeon*, emphasizing that recutting poor gears with a fine cut on a good machine was of less value than drastic hand finishing. The N.P.L. axial-pitch gauge was used during the recutting of the gears of *Sutlej* and *Jumna*, and it was found that gear surfaces would have to be much improved before the helical angles could be said to have been defined within 0.001 in. as measured along the helix. These cases threw further doubts on the provision and maintenance of alignment in the gearcase.

The effect of the redistribution of load caused by the torsional and transverse deflections was found to be closely connected with the incidence of failure. A small gear-set of standard Admiralty materials had successfully run at a $\frac{p}{d}$ value of 300 during tests carried out by Messrs. W. H. Allen, Sons and Company. Here, p is the tangential load per inch of tooth face width, and d is the pitch-circle diameter. Similar tests on automobile gears showed the ability of gearing to transmit very high loading and, aided by the marking experienced on gears, confirmed this analysis.

The first failure by breaking at the root occurred in the *Glasgow* in June, 1941. Prior to the failure, she had steamed on two shafts for several hours at one and one-quarter full-power torque, and on another occasion at one and one-tenth full-power torque, having lost the remaining two shafts by torpedo damage. In the next few months the low-pressure pinions were at full-power torque for over 6,000,000 revolutions, and on the next examination their teeth were found to be broken, and showed typical fatigue fractures. The wheels showed the markings of heavy daily bands. This failure led to an attempt to correlate quantitatively the redistribution of load caused by the deflection of the pinion by errors in helical angle, alignment, and by daily bands, with the results of fatigue tests on large numbers of gears.

The investigation of load redistribution showed that, when the change had been made for three- to two-bearing pinions, the criterion of total deflection calculated by rules then in use had failed to ensure the maintenance of the prevailing ratio of maximum to mean load. Less relief of misalignment was found to be given by two bearings than

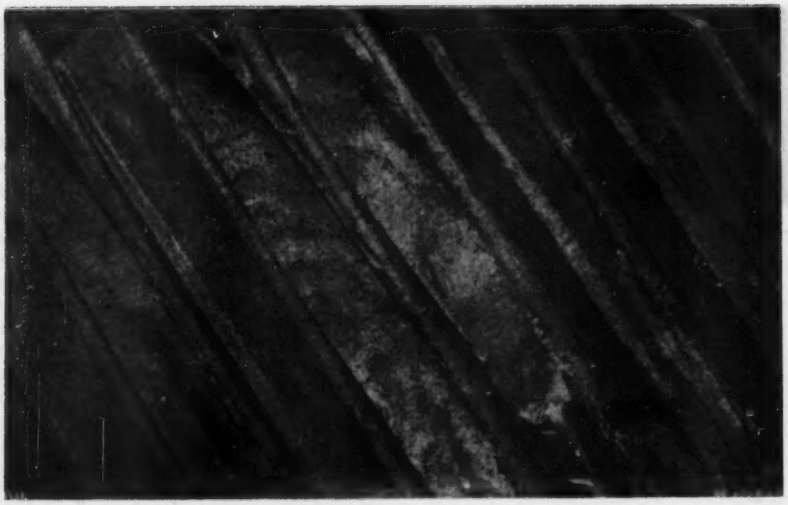
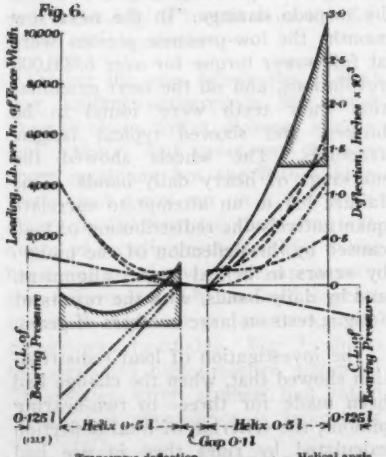


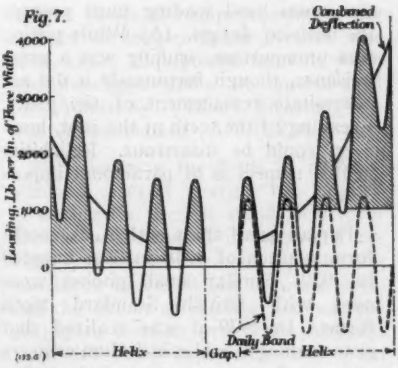
FIG. 5. FATIGUE FRACTURES; H.M.S. "FORMIDABLE."

Fig. 6.



Slew is the re-adjustment of the pinion axis caused by a change in oil-film thickness of bearings under unequal loads; *l* is the effective face width.

by three—because of the consequent change in bearing oil-film thickness. Service experience of gear marking had confirmed the existence of various possible load distributions, and it was now seen that, with a mean loading of 1000 lb. per inch and normal meshing, the maximum local loading could reach 3250 lb. to 4000 lb. per inch; with a helical-angle error of 0.0015 in. over the face width, 4200 lb. to 5500 lb. per inch; and with a daily band 0.0005 in. deep, 4900 to 6500 lb. per inch. The exact value depended on the value for the tooth stiffness, which, however, could not then be accurately estimated. Fig. 6 shows pinion deflection and loading curves with the effect of slew and helical-angle error, and Fig. 7 shows the effect of daily bands on load distribution, the pinion loading being represented by the shaded areas. Consideration of the effects of this redistribution of load, in conjunction with the results of fatigue tests, revealed



that the margin of safety was small, and depended on the careful hand work and meshing that had been carried out.

In all cases of failure the pinion steel was found to be of a suitable quality and micro-structure. The typical conchoidal marks of a fatigue fracture were visible with their origin at the roots of the teeth. No sign of hammering, bruising, or pitting of the working surface was visible. Fig. 5 shows the condition of the after end of the pinion of the center gear set of *Formidable* after failure.

The gear sets of *Strathmore* and *Griffin*, and the port-outer high-pressure set of *Ajax*, had V.B.B. teeth of 0.045-in. root radius, while other cases had normal semi-circular fillets. With the latter tooth-form the stress concentration is, say, 1.4, and in the former, 1.7. In gears with the V.B.B. tooth-form there is added uncertainty, due to the change in the load distribution up the length of the line of contact, with any change in center distance. Quite apart from manufacturing accuracy, this change in center distance may be influenced by temperature differences of the gearwheel and gearcase, and lead to unknown changes in maximum root stresses.

The case of *Formidable's* center shaft was confused by the contemporary incidence of axial shaft vibration, by a seized flexible coupling, and by a wiped forward pinion bearing. These factors did not arise on the starboard shaft. Initially all three of her gear sets apparently had good matching helical angles of pinion and wheel. Light was thrown on both failures by examination of the port gear set, and by fitting the spare center set, cut on an identical machine using the same feed. Some wear had occurred on the crests of the undulations of the port-set pinions. Where the polished patches had joined to give a continuous bearing surface, the hollows left by the hob were still apparent.

With the spare set, the helical angles of wheel and pinion, ahead and astern, roughly matched when the wheel and pinion were assembled in the gearcase in fine-clearance bearings. Marking was confined to circumferential bands on the after helix, covering less than 20 per cent of the face length, and on the forward, less than 30 per cent. After five weeks continuous filing of the wheel by shifts of fitters, this marking was extended to three-quarters of the face width. Circumferential bands of hollows with no contact remained. Figs. 8 and 9 show waves in the surface of the teeth of a pinion and main wheel, respectively of *Duke of York*, even where bearing has not been occurring. The waves could be readily felt by hand.

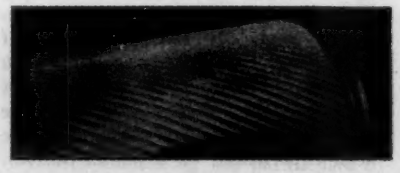


FIG. 8. WAVES IN PINION TEETH; H.M.S. "DUKE OF YORK."

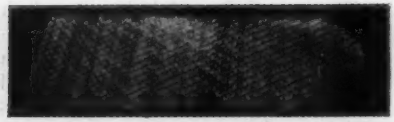


FIG. 9. WAVES IN WHEEL TEETH: H.M.S. "DUKE OF YORK."

In all these cases tooth failure has occurred along the pinion helix. Its position is determined by the concentration of load consequent on the deflection of the pinion under load, the daily bands due to lack of temperature control during machining, and the errors of helix angle of the gearing and of alignment in the gearcase bearings. Relief due to wear, of the bad tooth surfaces, did not occur in time to prevent failure with either A.-A. or V.B.B. teeth. Fig. 10 shows a tooth failure of H.M.S. *Ajax*.

The principal lessons learnt from war-time experience are as follow. (1) To keep noise and vibration, caused by periodic errors coming from the worm-table drives of non-creep machines, to a safe level, the depth of undulation from such errors must not exceed 0.0003 in. (2) To eliminate the risk of scuffing and pitting, load must be transmitted by at least two-thirds of the working surfaces of the gears. This may be ensured by keeping the surfaces free from undulations of more than 0.0001-in. range. (3) To eliminate the risk of scuffing, pitting, and breaking, the relative helical angles of pinion and wheel, the parallelism of their axes in the gearcase bearings, and the freedom from cross-wind of the gearcase, must be measured and controlled. (4) The effect of pinion deflections in concentrating the load at the outer ends of the helices was critical. Through lack of an earlier quantitative evaluation of this effect, the maximum loading at the ends had been higher than would have been experienced on a pinion of two-thirds the face width. The value of maximum local loading must provide the basis of design. (5) While pitting was unimportant, scuffing was a great nuisance, though fortunately it did not necessitate replacement of the gears. Breaking of the teeth at the root, however, could be disastrous. Reliability in this respect is of paramount importance.

For all small ships with A.-A. tooth-form, a pitch of 4/10 in. was adopted in 1939. Similar small pitches were used with British Standard tooth forms. In 1939 it was realized that gear-scuffing troubles in Italian cruisers was not due to excessive pitch, and a pitch of 8/10 in. was then adopted for the fast minelayers, *Ark Royal* (1942) and light fleet-carriers (1943). The designs of tooth-form adopted recently benefit from the experience of the United States Navy as well as of the Royal Navy. Pinion length: diameter ratios do not now exceed 2:1.

FIG. 10. TOOTH FAILURE: H.M.S. "AJAX."

The development of shaving existing gearing in place, will enable the present gearing of the existing large ships to be given a useful margin of safety against root failures, without incurring the expensive removal of the gears through an armored deck. Two sets of gearing of a destroyer have been shaved in this manner, the wheels in their gearcases in place in the ship, and the pinions in the shop. Although these gears were originally of a good

hobbed finish, the improvement in the amount of load-carrying surface was remarkable. Equipment for this work was developed under the auspices of the Admiralty Vickers Gearing Research Association (A.V.G.R.A.).

Experience in the fields of automobile and aircraft gearing has assisted in setting the present standard of production and resulted in improved reliability. Research to investigate the conditions in the oil film between teeth, and the effects of scuffing and surface pitting is now in progress. Already, from full-scale experience and from disc testing, it is clear that any undulation of less than ½-in. wave length, and more than 2×10^{-5} in. deep, leads to local areas of increased heat generation and of increased stress, and hence to a reduction in reliability.

The additional loads due to misalignment and to undulations of long wavelength can be roughly estimated. Where the combined deflection of the engaging teeth under a load of 1000 lb. per inch of face width is 0.0004 in., an error of 0.001 in. over the face in helical angle, or in parallelism of the shafts perpendicular to the common plane, will entail a difference in load between the ends of 2500 lb. per inch of face width, and an increase in the maximum loading by 1250 lb. per inch of face width. A long undulation due, for example, to a daily band of 0.0004-in. depth, will impose an increase in the maximum local loading of 500 lb. per inch. With a fine surface finish and the present soft materials, wear does not correct helical-angle errors, daily bands, or the strain due to pinion deflection under load. Stronger materials, in general, provide an increase in resistance to wear greater than the simultaneous increase in resistance to root failure. It is evident that economical use is not being made of gears until their alignment is within 0.0005 in.

To obtain the necessary standards of accuracy, essential requirements in gear cutting and finishing are: (1) Hobbing machines to B.S.S. 1498-1948, grade "A." (2) Temperature control of machine and cutting, to ± 1 deg. F. (3) A post-hobbing process, such as shaving or lapping. Such equipment is now in limited use. In some cases the permissible errors on the as-hobbed gears according to B.S.S. 1498-1948 have been halved. Shaving or lapping has then improved the surface and corrected slight differences in helical angle between the pinion and wheel. Grinding provides similar accuracy and finish. For this correction, and for a close control of helical angles, two further items are essential: the axial pitch gauge and an accurate meshing-frame. With this equipment and skilled craftsmen, it appears practicable to maintain a match of helical angles of pinion and wheel to within 0.0003 in. No quantitative control of profile on large gears is practiced at present.

Dimensional accuracy in the production of marine gears and their gearcases is of prime importance, and the achievement of such accuracy has been due, in the main, to the development of new measuring apparatus. The undulation recorder made a great improvement in quality of the product, and the axial-pitch gauge and meshing frame are making another. The development of means to assess the accuracy of alignment of gears and gearcase assemblies is still in the initial stages.

The present practice is that double-helical gears are hobbed, shaved, and meshed in a meshing frame. The pinions are selectively shaved to the wheel. Single-helical gears are ground, and meshed in a meshing frame. The pinion helical angle is reground to match the wheel: meanwhile, the gearcase has been bored. When mandrels, levels, and stick gauges are used, the error in measuring the parallelism of the bores may be up to 0.002 in. With

improved equipment this should be reduced to 0.001 in. The bearings are fitted to the gearcase and the gears placed in them. Marking is then applied to the gears and one of the pinion bearings is adjusted to give the best meshing, which will provide alignment within perhaps 0.0003 to 0.0005 in. over the face. This meshing test is at present the most accurate check of the parallelism of the axes of the gearing elements.

In the United States during the 1939-45 war, for a number of "locked-train" gearcases of similar design, jigs were in general use, and much setting-up time was thus saved, though the accuracy of parallelism of boring was no higher than with other methods. Two attempts are now being made to save setting-up time, and, in addition, to bore to a higher degree of accuracy with jigs. When jigs are used, the first essential is to set them up in the correct relative positions, free from cross-wind, and the second is to employ adequate steadies correctly aligned. The alignment of the steadies may be checked by means of an alignment telescope and targets, and after boring the results may be measured by the same means. A recent test showed that the tolerances required in locked-train gears, to ensure even sharing of the load between the trains ahead and astern, have been obtained, and that the accuracy of measurement of parallelism is about 0.0005 in.

As gears are produced to such high standards of accuracy, the possibility of achieving interchangeability has assumed importance. In the event of failure of the gearing, interchangeable parts may be replaced with no hand fitting. The alternative is to allow an adjustment of one of the bearings to provide true meshing, the adjustment being done preferably in a controlled manner using a fixture for boring, or by a pad and adjustable liner behind the shell as in the steam gun-boat pinion bearing. In either case, with double-helical gears, where it is desired to replace only one of a pair of meshing gears, provision must be made to ensure that the included helix angles of the original and replacement gears are held to close tolerances. The pressure angles and base pitches must be similarly controlled.

If conventional bearings are to be used, and no hand fitting is to be allowed, extremely close tolerances would have to be maintained on the difference in dimensions between the following features, in pairs, for any one pinion or wheel: journal diameters; bearing bores; bearing-shell thickness; bearing outside diameter; and bearing-housing diameters. All these tolerances, except for that of bearing-housing diameters, could be provided now, but the tolerance on the parallelism of the bores of the gearcase housing would have to be reduced to about one-third of the best achieved to date. For the present, therefore, adjustment of the bearings by hand must be accepted.

The freedom from cross-wind of the gearcase has been checked by the use of an alignment telescope on an accurate rotary table, the departures from a plane of targets on selected positions also being checked. It is not yet clear whether this method will control the cross-wind to the degree of accuracy required. A gearcase is being designed which will have the torsional stiffness requisite for mounting on three local areas. In view of the adverse working conditions under which the checking of gearcases is undertaken in war-time, the ability to dispense with the difficult check on cross-wind would be welcome.

In the field of small gearing, several methods of producing heat-treated steel gears are in use. Three of these methods may be used in the production of large marine gears, namely: (1) hobbing and shaving the already heat-

treated material. Shaving of material up to 80 tons per square inch ultimate tensile strength has been accomplished, though this may exceed the economic limit. (2) Cutting and grinding the already heat-treated material. Steel of 80 tons per square inch ultimate tensile strength, similar to En 30, has been used for gearwheel rims finished in this way. (3) Hobbing or cutting, case-hardening and grinding. Steel similar to En 36 has been used for 70-in. diameter gearwheel rims. Until recently, the use of a medium-carbon steel for gearwheels was justified on the grounds that wheel-teeth wear could relieve errors in alignment, or, alternatively, by local yielding at the root, avoid fatigue failure. Experience of bruised teeth, and knowledge of the increased yield strength of carbon steel at rapid rates of loading has shown that neither of these safeguards really exist.

The primary quality required of gear materials is resistance to failure in bending under the stress concentration to be found at the root, say, 1.3 to 1.7. Another desirable quality is resistance to pitting and wear. This latter is, in general, a measure of the hardness. With regard to resistance to fatigue in bending, data from full-size specimens are difficult to find, and tests are now going on by A.V.G.R.A. to provide further information on typical steels. As freedom from tooth failure by root fracture is the primary requisite, the justification for a difference of allowable loading between naval and merchant ship gears, based on difference of hours running at full power, disappears. Neither type of ship can afford such failures. A difference in standards of production would be the only possible justification for different loading standards.

An increase in root strength of hobbed and shaved gears may shortly be obtained by skin hardening through induction heating, using a steel of adequate carbon content. This process should not incur excessive distortion. It is not yet available for production, however. With small gears this method of surface hardening does not produce an increase in operating strength comparable with that obtained with the same increase of hardness in a through-hardened steel, but with large gears experience suggests that the converse is true. Increased interest in the processes of case-hardening and grinding resulted. For gearwheels too large in diameter for case-hardening, air-hardening steel could be used, but grinding would still be necessary.

The inherent possibilities and limitations of the two techniques, of hobbing and shaving heat-treated steel, and of case-hardening and grinding, have thus become prominent. It will be of value to examine two current designs of similar power using these two techniques, and loading based on existing experience, and the production development and testing work now being undertaken to maintain production standards.

Two alternative designs have been accepted for a ship in which the gearing is a significant part of the machinery weight, namely, double-helical, hobbed and shaved gears; and single-helical, case-hardened and ground gears. The first of these relies on fulfilment of the basic requirements of: close temperature control; reduction of undulations to a degree permitting rapid removal of material by shaving, and, or alternatively, lapping, and close control of helical angles; and safeguarding the gear meshing by adequate control of the boring and adjustment of gearcase housings and bearings. In addition, the first design is based on either the use of the strongest material that can be hobbed and shaved accurately, and produced in the quantities required in an emergency, or the use of

a slightly inferior but more readily available material necessitating correction of the helical angle of the pinion to provide even distribution of loading at high powers.

The loading chosen is based on experience with these materials and on the results of the United States National Boiler and Turbine Laboratory back-to-back tests. With the second design, the material of the pinions and wheel rims of the single-helical gear sets will be similar to specification En 36, but will have a low carbon content. Both designs are required to fit the same shaft and turbine centers. No correction of the pinion for deflection is envisaged for the second design.

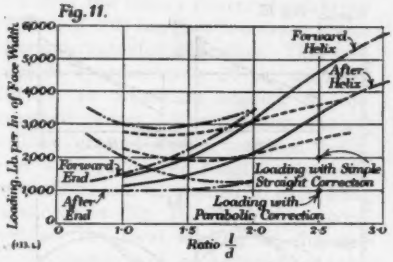
The following are details of materials accepted at present:

Type of Gear	*Specification of Material*	*Composition of Material*	*Ultimate Tensile Strength, Tons per sq. in.*
Primary and main wheels	En 30	0.3 per cent carbon; 4¼ per cent nickel-chromium	60-65
Primary and secondary pinions	En 26	0.4 per cent carbon; 2.5 per cent nickel-chromium-molybdenum	70-75

A great deal of information on the aspects of design and production standards which control the load-carrying capacity of gearing can be gained from back-to-back testing on shore, and accordingly tests have been arranged of hobbed and shaved gears, to be carried out by *Pametrada*, using: (*a*) spare gearing made for the steam gunboats; (*b*) gearing of harder materials in the same gearcases; and (*c*) gearing with pinions corrected to compensate for deflection under load. Sets (*b*) and (*c*) will be produced by A.V.G.R.A. Tests of case-hardened and ground gears are to be carried out by *Pametrada* using gears produced by A.V.G.R.A., representing a primary train of a double-reduction set. Similar tests are to be carried out by Messrs. Vickers-Armstrongs Limited, Barrow, using gears produced by A.V.G.R.A. and ground by the Maag Gearwheel Company of Zurich, representing the secondary train. A second set of test gears has been made by Maag employing steel supplied by Poldi, which they are accustomed to using.

The limitations on the length: diameter ratios of pinions caused by deflection are illustrated in Fig. 11. This diagram has been drawn for only one value of tooth stiffness and bearing size, but its general implications apply to pinions as a whole; it shows the load concentration on single and double helical pinions. With an $\frac{l}{d}$ ratio of 2.5, where l is the effective face width and d the pitch-circle diameter, the maximum loading, with simple straight correction would be 2000 lb. per inch, and with parabolic correction, 1000 lb.

Fig. 11.



Values of maximum local load with :—

Total load equal to that with mean load of 1,000 lb. per inch on $\frac{l}{d}$ of 2·0 { — · — · — Single helical. — — — — Double helical.

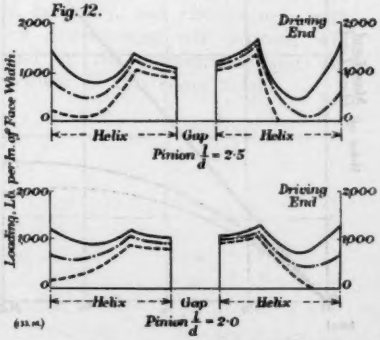
Mean load of 1,000 lb. per inch { — · — · — Single helical. ———— Double helical.

per inch. With the same $\frac{l}{d}$ ratio, the maximum loading for uncorrected pinions would be 4500 and 3300 lb. per inch on the forward and after helices respectively. Ideally, the helix angle should be continuously varied over the whole width, this being referred to as parabolic correction. An approximation to this may be obtained by two or more steps in each of which the helical angle is kept constant; this is known as straight correction.

The idea of correcting pinion helices was considered by Parsons, but gears are only now being made of sufficient accuracy to permit the application and measurement of these corrections. While correction for deflection and misalignment is common practice in the automobile and traction fields, its application to the marine field has been delayed, not merely because of practical difficulties, but also because of a number of uncertainties. These were: the limits within which the position of the center of the bearing reaction is known, and hence, in part, the amount of deflection imposed; the limits within which the value of tooth stiffness is known; the effect of wear at partial loads; the problems of meshing; and the lack of actual practical experience. An adequate estimate of the position of the center of the bearing reaction can be made by assuming that the pressure carried by the oil film is inversely proportional to its thickness. Departures from parallelism of the journal and bearing axes shift the center of pressure less than one-fifth of the bearing length. An estimate of tooth stiffness can be made from available information. Wear at partial loads does not affect the correction given to traction and automobile gears of hard material. The use of a dummy pinion, with teeth made to the unloaded and uncorrected helix angle, to be used against the wheel in the meshing frame and in the gearcase bearing, should enable satisfactory meshing and alignment to be maintained. The necessary tests are now being undertaken.

The simplest form of correction of double-helical gears is to hob or grind the pinion to the helix angle which will give similar loading at each end of the helix under load. An improved form is to provide the unloaded helix angle on the inner thirds of each helix and to apply the corrected helix to the outer two-thirds only, as shown in Fig. 12.

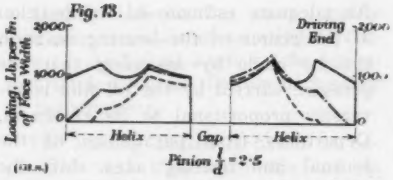
Fig. 12.



Full load. ———— Three-quarter load. — — — — Half load.

Corrected over two-thirds of each helix.

Fig. 13.

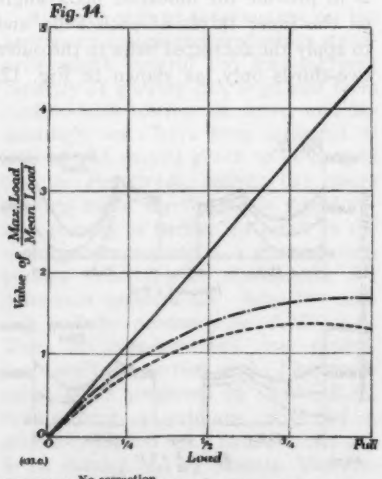


——— Full load. — · — · — Three-quarter load.
— — — — Half load.

Corrected over two-thirds of each helix.

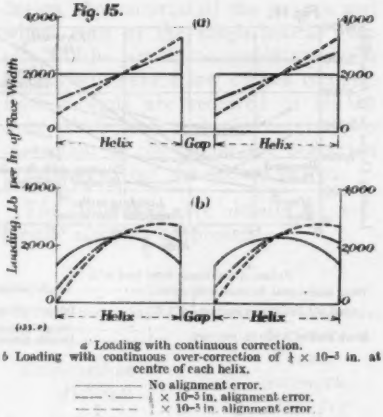
Shaving back the outer quarter gives further improvement, as shown in Figs. 13 and 14, and safeguards against high end-loading due to small errors in alignment. It is hoped to apply this kind of correction to the gears of existing ships when the wheels are shaved. One of the sets to be tested by *Pametrada* will have a correction applied in this way; the other will have a continuous correction over the whole face. This has been provided by using correcting-cam control of the hob saddle-nut as used in master screw-

Fig. 14.



——— No correction.
— · — · — Corrected over two-thirds of each helix.
— — — — Corrected over two-thirds with ends shaved in addition.

Fig. 15.



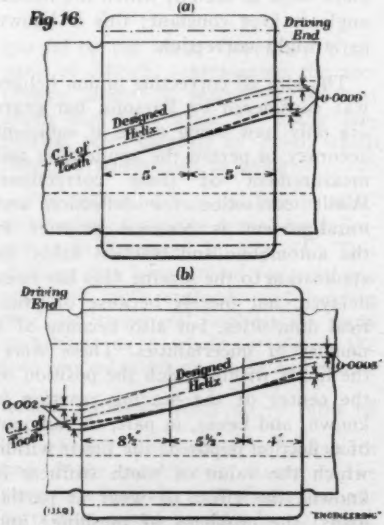
a Loading with continuous correction.
b Loading with continuous over-correction of $\frac{1}{2} \times 10^{-3}$ in. at centre of each helix.

——— No alignment error.
— · — · — $\frac{1}{2} \times 10^{-3}$ in. alignment error.
— — — — $\frac{1}{4} \times 10^{-3}$ in. alignment error.

cutting lathes. A slight over-correction to safeguard against alignment errors can be provided with advantage, as indicated in Fig. 15.

The test pinions will be ground over half their faces to the nominal helix,

Fig. 16.



a Test pinion (second reduction).
b Loading pinion (second reduction).

and over the halves nearer the driving ends they will have an increased angle to compensate for deflection. This will provide approximately 20 per cent reduction in maximum loading at the designed power. The loading pinion will be ground to the nominal helix over a third of the length only, and corrections will be applied to each end, as shown in Fig. 16. With these corrections the local loading at the designed power will be reduced to half that otherwise imposed. Without them it would not be certain that this pinion would outlast the test pinions and, hence, be available for later testing of other sets.

For second-reduction gears an investigation was made into the amount of tip relief required to relieve the stress concentrations due to the "digging-in" effect at the roots of the mating teeth, the buttress effect of the adjacent unloading tooth section, and the improvement of local lubrication conditions; the tip relief required at 24,000 h.p. was estimated to be 0.002 in. Provision has also been made for end relief of 0.003 in. over a distance of ½ in., for both the secondary and primary gear-trains. In both cases the roots of the teeth are to be polished in a direction perpendicular to the groove, to improve fatigue strength.

Two additional tests will be carried out in the second-reduction series. In one, both pinion and wheel will be made of steel to specification En 24, and skin hardened by induction heating, and, in the other, a case-hardened and ground pinion will drive an air-hardened hobbed and shaved wheel with ultimate tensile strength of 60 to 65 tons per square inch. The outcome of these tests will not be known for some time, but the following implications of the two designs may be noted. Serving as an articulated design for two turbines at their present accepted loadings, they could transmit 5800 hp. to a shaft running at 100 rpm., from turbines running at 2500 rpm. With the same diameters, and face widths increased to give length : diameter ratios of 1.8 and 2.6 for the single and double helical gears respectively, and if the tests confirm that the corrections for deflection are fully effective, they could transmit 10,000 hp. The amount of additional power, which the tests confirm can be carried safely, will be of great interest.

While a great deal of work has been carried out in the past few years in improving marine gears, much remains to be done. As in other fields of engineering, the ideas which collectively provide the thread of development, and which have come to many different persons in various countries, have, in their translation into practice made demands on the techniques of various branches of engineering. The future pace, certainty, and effectiveness with which development will proceed will depend on the continuance and improvement of this cooperation.

TECHNICAL PROGRESS IN MARINE ENGINEERING DURING 1950

ACKNOWLEDGMENT

This article, which was published in the January 1951 issue of "The Shipbuilder and Marine Engine-Builder," deals principally with developments in Great Britain in marine engineering during 1950.

Over the years, the reviewer of technical progress in marine engineering has invariably been taxed to find a development of the previous 12 months which could, in truth, be described as the outstanding one of the year. Occasionally there has been a landmark, such as the adoption of high-pressure, high-temperature steam machinery, the introduction of pulverized fuel for marine use, the coming of the double-acting marine Diesel engine, and so on. Looking back, such years have not necessarily been the most fruitful in point of sound and sustained progress, even if they have provided an acceptable landmark around which to build this yearly review. The years in which high activity and steady progress have been the order of things are inevitably the most fruitful years from the point of view of marine engineering progress; which is as it should be.

DIESEL ENGINES

Perhaps the most significant development of the past 12 months has been the success which has attended the employment of heavy fuel oil, sometimes referred to as boiler oil, in Diesel propelling machinery. Without exploring the economics of these heavier grades of oils as fuels in motorships, it is now clear that their utilization by motorship owners is justified and has come to stay.

In this connection, the work done by Mr. John Lamb, O.B.E., of the Anglo-Saxon interests, is outstanding. His patient researches in association with Messrs. R. and W. Hawthorn, Leslie & Co., Ltd., St. Peter's, Newcastle-on-Tyne, culminating in the employment of heavier fuel oils (named Ordoil by the Shell organization), are fresh in memory. The *Auricula*, which has now been in service for a considerable period, has amply justified Mr. Lamb's confidence in this important development. Other ships of his company's fleet have confirmed these initial successes. The British Tanker interests and Messrs. William Doxford & Sons, Ltd., have also done important work in this field during the past year or so, and other well-known British motorship owners are successfully running vessels, many with Doxford and Harland-B. & W. machinery, in which heavier grades of fuel oil are employed.

It is fitting that tribute be paid to the close cooperation of the leading manufacturers of centrifugal purifiers. Without suitable purifying and clarifying machinery, the use of heavy fuels would not have been attended with the good results so far realized. Indeed, so successful has been the whole broad development, that even heavier and more difficult fuels are likely to be used in the future, and there is no doubt that the employment of so-called boiler oils in the auxiliary engines of motorships is the next forward step in this interesting connection.

On the subject of motorship progress generally, one of the significant developments of recent years has been the rising popularity of the opposed-piston engine. We have previously commented upon the marked preference which most superintendent engineers and operators have for the single-acting engine, largely on grounds of ease of maintenance. Despite the increased number of bearings and the more expensive crankshaft of the opposed-piston engine, this design is, at the present time, the type of single-acting engine most favored for installations of any considerable power. The Doxford opposed-piston design is today built under license by more than 20 firms in several countries and, during the past year, further license arrangements were concluded—the engine is now, for example, also to be built in Norway.

The first of the Doxford trawler-type engines went into service last year in the single-screw vessel *Lammermuir*. As the description of the unit at the time of its completion showed, this engine is essentially a scaled-down version of the larger Doxford engines, Messrs. William Doxford & Sons, Ltd., having wisely refrained from introducing any untried features. It is understood that a number of trawler owners are interested in this type and, we believe other engines of this design will be put in hand in the engine works of Messrs. John Lewis and Sons, Ltd., Aberdeen, in the near future. It is our opinion, however, that this new development, interesting though it is, is not likely to be so important in the future as the present tendency towards even more powerful versions of the Doxford engine. In this connection, it may be recalled that the parent firm have constructed a large single-cylinder experimental engine at their Palmers Hill Works at Sunderland, and even larger units are under consideration.

Meanwhile, Messrs. Barclay, Curle & Co., Ltd., Whiteinch, Glasgow, have recently completed the most powerful Doxford engine which has yet been built, although the cylinder dimensions are not so large as those of some Sun-Doxford engines built a few years ago. This Barclay Curle-Doxford six-cylinder engine has a bore of 750 mm. and a combined stroke of 2500 mm. (1450 mm. + 1050 mm.), and the normal designed output is 8000 B.H.P. at 104 rpm. The engine is one of a pair ordered by Norwegian owners for single-screw oil-tank ships of 18,500 tons deadweight, and several other engines of this size are on order. The engine is of the now familiar type, with three lever-driven scavenging-air pumps, and a Doxford-Bibby detuner is fitted to the forward end of the crankshaft. It is interesting to note that, on test, 9068 I.H.P. was carried for one hour at 104 rpm., representing 8000 B.H.P., i.e., a mechanical efficiency of 88.3 per cent.

Earlier last year, equally interesting two-stroke cycle, opposed-piston engines of the Harland-B. & W. design were built at Belfast and Greenock by Messrs. Harland and Wolff, Ltd., and Messrs. John G. Kincaid & Co., Ltd., respectively, for installation in single-screw vessels. These units, it is of interest to note, have the same cylinder bore as the Doxford unit just referred

to, while the main-piston stroke is 1500 mm. and that of the exhaust pistons 500 mm. The Kincaid-built engine was the first of a series ordered by Messrs. Alfred Holt & Company, and was installed in their single-screw motorship *Bellerophon*. The maximum service rating is 7600 B.H.P. at 110 rpm. and it is arranged to run on heavy fuel oil.

In general design, these Harland-B. & W. opposed-piston engines (sometimes referred to as the "coverless" type) differ basically from the Doxford conception of the two-stroke cycle, opposed-piston engine. The design which Messrs. Harland & Wolff have developed during recent years is supplementary to their double-acting engine, and makes use of the well-tried arrangement of side rods and eccentric rods, the upper-piston eccentrics being integrally cast with the cast-steel main crank webs. This construction, of course, makes for a fairly short crankshaft. The layout works well in practice and results in a compact and accessible engine. Unlike the Doxford unit, scavenging air is supplied by twin rotary-type blowers.

In regard to performance, maneuverability and flexibility, there seems to be nothing to choose between the two types. Both are good examples of the large directly-coupled marine oil engine which, despite the alternative forms of indirect drive, is likely to be the principal type of motorship machinery for many years to come. Indeed, it would not be indulging in overventuresome prophecy to say that these large single-acting, directly-coupled oil engines by no means represent the limit of power output which we are likely to see in single-screw motorships. Steam machinery for such vessels has stepped ahead of the Diesel engine, with present shaft outputs of 14,000 to 15,000 H.P., but it is unlikely that the protagonists of the motorship will remain satisfied with outputs of the order of 8000 S.H.P.

A new two-stroke cycle engine of neat design and considerable promise is the Werkspoor-Lugt unit, the development of which has gone steadily forward since the death of Mr. G. J. Lugt. This engine, with its ingenious valve gear, excellent accessibility and arrangements for supercharging, is of an advanced design which should be simple and economical to build. Up to the present, however, there has been no announcement of orders for such engines having been placed.

INDIRECT DRIVE

There were indications last year that interest in indirect drive for motorships was everywhere increasing. In the December Number of this journal, for example, a description was given of the main machinery of the oil-tank motorship *Atlantic Duchess*, which is propelled by two Gray-Polar two-stroke cycle, six-cylinder engines driving a single screw through Vulcan hydraulic couplings and David Brown single-reduction single-helical gearing. The machinery arrangement is neat and of moderate weight, and the engines require little headroom—features which should commend themselves particularly to owners seeking an attractive layout for passenger motorships. The total output of these engines is 4720 B.H.P. at 250 rpm., which speed is reduced to 120 rpm. at the propeller shaft. The engine installed in the *Atlantic Duchess* and in other vessels having Gray-Polar installations is the latest larger Polar MT-type unit, which develops approximately 400 B.H.P. in a cylinder 500 mm. in diameter by 700 mm. stroke. The scavenging arrangements are interesting in that twin rotary blowers of simple design are provided, one for ahead operation and the other for astern.

Engines of the same general type are also manufactured by Messrs. John G. Kincaid & Co., Ltd., but, up to the present, they have not been built in the Govan works of British Polar Engines, Ltd., an establishment which, incidentally, is remarkably busy at the present time and which has a very full order book for the future.

Another medium-power, two-stroke cycle, all-port engine which has proved itself in marine service is the Crossley so-called scavenge-pump, exhaust-pulse-charging design, which is noteworthy for the close attention which has been given to the cylinder-scavenging and charging processes. One of the most interesting Crossley installations of the past year was that fitted in the Swedish-built single-screw motorship *Etrog*. In this vessel (noteworthy as the most highly-powered Crossley-engined vessel yet completed), two Crossley standard reversible two-stroke cycle engines drive a single screw through electromagnetic couplings and single-reduction gearing. The two engines are rated to develop 3000 B.H.P. at 320 rpm.; the propeller shaft rotates at approximately 124 rpm.

The name of Sulzer Brothers has, of course, been identified with geared Diesel drive in many important installations, the most noteworthy being that of the Rotterdam Lloyd motor liner *Willem Ruys*, of 32,800 B.H.P. total output. Recently, orders have been placed by the Compagnie Générale Transatlantique and the New Zealand Shipping Co., Ltd., for geared installations having the latest type of Sulzer two-stroke cycle engine. Four orders have been placed by the latter owners, all with geared Diesel engines of the Sulzer type. Messrs Alexander Stephen & Sons, Ltd., Linthouse, Govan, Glasgow, are to build three. The lowest-powered vessel, to be named the *Cornwall*, will have two Stephen-Sulzer eight-cylinder engines, each of 4000 B.H.P. and transmitting their power to a single screw through Vulcan-type hydraulic couplings and single-reduction gearing. The main engine rotates at 225 rpm. and the propeller at 100 rpm. The second vessel, to be named the *Surrey*, will have two similar nine-cylinder engines, each rated to develop 4500 B.H.P. at 225 rpm. In this installation, the drive to the gearbox will be taken through B.T.H. magnetic couplings, and here, again, the propeller speed will be 100 rpm. A similar but more powerful layout is to be employed in a third vessel, which will have two Stephen-Sulzer ten-cylinder engines. The fourth of the series is to have Sulzer engines manufactured by Messrs. John Brown & Co., Ltd., Clydebank, and each engine will be of 5500 B.H.P. An interesting feature of all these important new single-screw vessels is that the engines are to be arranged for operation on heavier-grade fuel oil.

Some time ago, Messrs. George Clark (1938), Ltd., organized their works at Sunderland for the construction of Sulzer two-stroke cycle engines of medium power. The test-bed trials of the first Clark-Sulzer engine was witnessed by representatives of the technical Press and created a most favorable impression. The unit has a service rating of 1125 B.H.P. and is a direct-reversing unit, with a separate scavenging-air pump for each cylinder, in accordance with the latest Sulzer practice. The engine, which was installed in the collier *Portsmouth* (2500 tons deadweight), was opened up after the vessel's first six months of service. During this period she had covered 16,000 sea miles and carried a total cargo of about 52,000 tons, with an average speed of 10.53 knots during loaded southbound trips. The corresponding propeller speed was 211 rpm. and the all-purpose fuel consumption over the period worked out at no more than 4.26 tons per day.

When the main engine was inspected, all eight pistons, liners, piston rings,

gudgeon pins, big-end bearings, etc., were found to be in excellent condition, while selected main bearings and other parts were opened up for inspection with equally gratifying results.

Other engines of this type are in hand, and there is no doubt that this well-designed engine represents a welcome addition to the range of medium-speed, two-stroke cycle marine engines of moderate power, available in this country.

Diesel-electric drive—the alternative form of indirect Diesel propulsion—has not been represented by many important installations during the past year. In this respect, mention must be made of the four important Argentine dredgers with English Electric pressure-charged Vee-type engines, which were followed by a vessel of the same general type from Holland, but with M.A.N.-type engines. This latter vessel and one of the Clyde-built craft are somewhat larger than the remainder; all were placed in service some time ago.

An interesting Diesel-electric vessel of the year is the pilot craft *Sir Thomas Brocklebank*, which Messrs. Philip & Son, Ltd., Dartmouth, have recently completed for the Mersey Docks and Harbor Board. This 13½ knot, 165-ft. vessel has a single screw which absorbs 920 S.H.P. at 300 rpm. The main machinery contractors were the General Electric Company, Ltd., of London, and the two Diesel engines which drive the 375-kw. main generators and 80-kw. tandem auxiliary generators are four-stroke cycle, six-cylinder units, supplied by the National Gas & Oil Engine Co., Ltd., Ashton-under-Lyne. Each prime mover develops 640 B.H.P. at 520 rpm. and pressure-charging is accomplished by blowers, driven by exhaust-gas turbines. The engines are of the latest National type, each cylinder having four valves actuated by a very neat valve gear.

This type of machinery is ideally suited to the conditions encountered in pilot service, and the performance of the *Sir Thomas Brocklebank* will be watched with keen interest by other harbor authorities. This compact and efficient machinery layout should appeal to overseas harbor authorities looking for an economical installation for craft of this type. The machinery has the important merit that it affords easy and well-nigh perfect maneuverability.

GAS TURBINES

In these yearly reviews, caution has been the watchword in the reviewer's comments upon the marine gas turbine. No subject is of greater interest to the marine engineer, and its potentialities for certain types of ships and services are unquestionably great. Some of the most eminent firms in the marine engineering world, including, among others, the British Thomson-Houston Co., Ltd., Messrs. John Brown & Co., Ltd., Messrs. Brown, Boveri & Co., Ltd., the Escher, Wyss Engineering Works, Ltd., the Metropolitan-Vickers Electrical Co., Ltd., Messrs. Ruston & Hornsby, Ltd., and Messrs. Sulzer Brothers in Europe, as well as several leading manufacturers in the United States of America, are giving the problems of the marine gas turbine deep study, most of them have had pioneer units on test or are approaching that stage. Pametrada—the research organization of the British marine turbine-building industry—is also closely engaged on marine gas-turbine work and has a number of advanced proposals under consideration.

The fact remains, however, that no large-scale marine gas turbine is yet at

sea. Dr. T. W. F. Brown, Director of Research of Pametrada, and other authorities in this field have made some interesting pronouncements in regard to the marine gas turbine, and not a few shipowners are sympathetically interested in development work. Now that boiler oil is being widely used as fuel for motorships, marine gas turbines will have to be capable of successfully using the heavier grades of bunker oil. This should not present any real difficulties for designers, and perhaps, before the next review of marine engineering progress is written, something further will be said on the general subject.

It should be recorded that a small marine gas-turbine installation was demonstrated a short time ago in the River Thames by the manufacturers of the Rover car, who have already attracted attention by their successful demonstration of a gas-turbine motor-car. Briefly, it may be recalled, the Rover Company installed, in a former Air Sea Rescue 60-ft. vessel, two of their small gas-turbine units for driving the wing shafts, the central screw having a Dorman Diesel engine for cruising at moderate speeds. Each gas-turbine unit in this craft comprises a compressor (driven by a gas turbine), while the output turbine drives the corresponding shaft, through 7:1 reverse-reduction gear, at a propeller speed of about 850 rpm. The vessel ran well, and the combustion characteristics were excellent, but the omission of a heat-exchanger from the circuit involved a penalty in high fuel consumption—as much as 1.25 lb. per B.H.P. per hour at the full output of 150 B.H.P., with ordinary Diesel oil. With suitable heat-exchanging equipment, there appears to be nothing to prevent an estimated specific consumption of 0.8 lb. per B.H.P. per hour being attained.

The weight of these turbines is very moderate, as are the overall dimensions of the plant; and, provided compact heat exchangers can be devised, a gas-turbine plant of this type, but of appreciably higher power, should have possibilities for the propulsion of motor torpedo-boats and other high-speed craft.

STEAM MACHINERY

In the steam machinery field, some interesting installations were completed during the year, and a number of these incorporate features of considerable technical importance. Although not by any means the largest installation of the year, the turbine machinery of the cross-Channel steamship *Brighton* comes immediately to mind. It may be recalled that the *Brighton* has two sets of *single-casing,* single-reduction geared turbines, in the design of which the Parsons and Marine Engineering Turbine Research and Development Association collaborated; like the ship, the turbines were built by Messrs. William Denny & Brothers, Ltd., Dumbarton. The design is interesting in that the split cast-steel turbine-cylinder barrel is carried within an outer shell by two pairs of palms at the horizontal center-line level, one pair being in line with the nozzle belts and the other at the after (or exhaust) end of the ahead portion.

The exhaust steam from both the ahead and astern sections passes through the cylinder barrel proper into the fabricated supporting outer casing, and is carried thence to the underhung condensers, which are of the Weir regenerative single-flow pattern.

The design possesses many ingenious and interesting features which cannot readily be discussed without a drawing. One of the most unusual of these is the elimination of insulation and lagging on the turbine casing, which remains cool at

all power outputs. The design is characterized by simplicity, and close attention has been paid to the avoidance of distortion. As the turbine is of all-impulse pattern, no dummy piston is necessary; any axial thrust is taken care of by the Michell-type turbine adjusting block.

The reduction gearing is interesting in that the all-addendum double-helical gears were shaved after being hobbed, in the interests of accuracy and silent running. Naturally, with a single-casing arrangement, the appearance of the sets is somewhat unusual; but, for a vessel of this class, the decision to adopt turbines of this type, wherein ultimate in economy is not so important as compactness, moderate weight, simplicity and high astern power, has been justified on all counts. The performance has been up to expectations and it is not unlikely that this installation will form a prototype for future British cross-Channel turbine steamships.

A very interesting machinery contract was placed last year by Messrs. Alfred Holt & Company, for two sets of single-screw geared-turbine machinery for vessels ordered from the Caledon Shipbuilding & Engineering Co., Ltd., Dundee. These geared turbines are designed to operate with steam at a pressure of 600 lb. per sq. in. (gauge) and a final temperature of 950 deg. F. Constructed by the Metropolitan-Vickers Electrical Co., Ltd., of Trafford Park, Manchester, these sets are designed for a maximum output of 8000 S.H.P. at 112 rpm., with an economical power of 7250 S.H.P.

Very close attention has clearly been paid to obtaining a high degree of economy, with a flat fuel consumption curve, and features which might be regarded as experimental have been excluded. A three-casing layout has been adopted, and only the low-pressure cylinder contains astern elements—the astern power is less than 45 per cent of the full ahead power, a figure which is considered adequate for a cargo liner. The h.-p. rotor is designed to run at 6000 rpm. at maximum output, the i.-p. turbine at 4500 rpm. and the l.-p. at 3000 rpm. The forged molybdenum-steel h.-p. rotor has integral wheels, and the i.-p. turbine wheels are also forged integrally with the shaft. For the l.-p. rotor, separate forged wheels are shrunk and keyed to the shaft, in the familiar fashion. Because of the high initial temperature of the steam, the h.-p. casing takes the form of a molybdenum vanadium-steel casting.

Steam is admitted to the h.-p. turbine via three nozzle groups, controlled from a single shaft by a valve which operates the nozzles in correct sequence, the high normal output of 7250 S.H.P. being obtained with two groups of nozzles. The double-reduction gearing is arranged in two separate cases of fabricated construction.

This interesting turbine design, which should show excellent economy, has no doubt been influenced by the Manchester firm's successful experience in the design and construction of geared turbines for oil-tank ships and cargo liners during the war, and its performance will be watched with great interest in due course.

One of the most important ships of the year, and one possessing a very interesting machinery layout, is the Anglo-Saxon Company's oil-tank ship *Velutina* (28,330 tons deadweight) on a draught of no less than 34 ft. 2 in. Built by Messrs. Swan, Hunter & Wigham Richardson, Ltd., this noteworthy ship has attracted considerable attention in that she is the first of the British so-called super-tankers—vessels which, however, will be dwarfed in size, if American plans to build steam-propelled oil-tank ships of no less than 40,000 tons deadweight materialize.

The *Velutina* has a three-casing set of double-reduction geared turbines de-

veloping 13,000 S.H.P. at 106 rpm. and designed by Pametrada. The three main boilers are of the Babcock & Wilcox single-pass water-tube type, and are neatly arranged in a compact boiler-room located above and aft of the main machinery. Two boilers are situated over the thrust recess, the third being on the same flat, immediately astern. Steam is generated at a pressure of 500 lb. per sq. in. and a temperature of 750 deg. F. The installation has been constructed by the Wallsend Slipway & Engineering Co., Ltd.

The double-reduction gearing is of the popular articulated (or cascade) type, whereas in certain earlier British single-screw cargo liners, with turbines of approximately the same power, the older form of interleaved gearing has been employed. An important feature of the boiler plant is the automatic combustion-control equipment—a development which, it should be remarked, is becoming increasingly popular in this country. It is certainly one which does not lack justification from the point of view of fuel economy.

Reference to automatic combustion-control equipment and appreciation of some of the design features being incorporated in modern British geared-turbine plant, make it fitting to observe that our American friends have, in recent years, made notable contributions to marine engineering progress. In addition to pioneering automatic combustion-control apparatus in ships, American manufacturers have, through their enterprise, increased the popularity of the two-casing geared turbine and the cascade form of double-reduction gearing, while the Foster Wheeler water-tube boiler, which has found much favor in this country, is an American design which also comes to mind in this connection. It is interesting to find, therefore, that two-casing geared-turbine machinery of American design and manufacture was last year installed in important Dutch tonnage, and is, in point of fact, to be fitted in some ships ordered a short time ago from Scottish builders.

The machinery installation of the Dutch single-screw liner *Diemerdijk* is representative of this class of geared-turbine equipment, consisting essentially of the standard General Electric machinery built for the American C3-class vessels. The normal rated output of 8500 S.H.P. is developed at 85 rpm. of the propeller, and steam is provided at a pressure of 440 lb. per sq. in. and a temperature of 740 deg. F. The two Foster Wheeler D-type boilers are equipped with Todd oil-burning apparatus and automatic combustion-control equipment.

The two-casing cross-compound turbines have astern elements in the l.-p. casing only. The two-stage velocity-compounded astern turbine, which is located at the forward end of the l.-p. casing, is capable of developing 80 per cent of the normal ahead torque at about 42.5 rpm., with full-ahead steam flow.

The double-reduction gearing is arranged in three planes, the axes of the two high-speed pinions being in the upper plane, those of the high-speed gear wheels and low-speed pinions in the intermediate plane, and that of the main gear wheel below, in line with the propeller shafting. The low-speed pinions have a quill-drive arrangement.

An interesting feature of the engine-room of this Dutch vessel is the American Aldrich variable-stroke reciprocating-type feed pump—a motor-driven unit, with plungers driven from the pump crankshaft by means of connecting rods with link-motion interposed. The discharge from the pump is varied through the servo-operated link-motion by means of a thermostat on the automatic feed-water regulator. The driving motor is a constant-speed machine, while the stroke of the three pump

plungers may be varied from zero to 4½ in., in the manner mentioned.

Reference has been made in the technical and shipping press during the past year to the fact that most of the reconditioning program carried out in British yards since the war is coming to an end. Last year saw the reintroduction into commercial service of three especially interesting large ships in which large-scale work has been carried out. The most important of these is, of course, the French liner *Liberté*, formerly the German liner *Europa*, which has been extensively refitted at the Penhoët yard, Saint Nazaire.

By current standards, the machinery layout of the ex-*Europa* is not modern, and, not unnaturally, the question of providing new turbines and boilers was not neglected. In the event, however, the existing machinery has been thoroughly overhauled; and, while the rather wasteful steam-driven auxiliaries have had to be retained, economy has been improved by replacing the original turbo-generators by modern machines. These sets are installed in one generator room, and the Diesel-driven equipment is in another. The four fairly large Diesel generators (with M.A.N. engines) have been replaced by modern dynamos, driven by General Motors Vee-type, two-stroke cycle, uniflow-scavenging engines, and these should prove to be more easily maintained at this stage of the vessel's career, and should be wholly reliable.

In pre-war days, the German East Africa 18-knot turbine ships *Pretoria* and *Windhuk* created something of a stir in technical circles, with their (for that time) advanced turbines and Benson boilers, designed for an initial steam pressure of 1135 lb. per sq. in. and a temperature of 896 deg. F. at the superheater outlet. Bold in conception, these installations were provided with Askania automatic equipment for controlling the flow of feed water, fuel oil and combustion air. In service, however, the ships were not entirely successful; and, like some of the German warships (also with Askania gear) of which great things were expected, performance in respect of fuel consumption and general reliability left something to be desired.

The *Pretoria*, after extensive modification by Messrs. John I. Thornycroft & Co., Ltd., at Southampton, is now the British troopship *Empire Orwell*. The two Benson boilers and Askania gear have been removed, and in their place a pair of Thornycroft-built Foster Wheeler controlled-superheat boilers have been installed. These are provided with the excellent Thornycroft oil-burning equipment of latest type, and are designed for a pressure and temperature at the superheater outlet of 500 lb. per sq. in. and 800 deg. F. respectively. The output of the turbines has been reduced from 13,000 to 11,000 S.H.P., with a corresponding decrease in speed from 18 to 15 knots. The three-casing turbines have been modified, the Pametrada Research Station having collaborated in planning the important changes which have been made in the interests of economy.

New steam piping has also been provided, and the large number of turbo-driven auxiliaries has been drastically reduced in favor of modern electrically-driven equipment. The German auxiliary turbo-generators have been overhauled, and the original geared turbines which drive the main circulating pumps have been modified in the light of the altered steam conditions.

Altogether, the difficult task of modifying and improving the machinery has been accomplished in a most praiseworthy fashion. Reliability should be normal, and the fuel rate at all loads is now appreciably better than it was with the original—and unjustified—steam conditions.

The other notable steam-turbine modification which calls for mention here is that which Messrs. Thornycroft have carried out in the renowned British liner *Monarch of Bermuda,* re-named the *New Australia.* The vessel now operates in the emigrant service between the United Kingdom and Australia. In the course of the extensive alterations which have been made, the appearance of the ship has been completely altered, though the G.E.C. turbo-electric machinery remains largely as originally installed. It has, of course, been completely overhauled, as have the four Babcock & Wilcox three-pass boilers. New four-blade propellers have been provided by the Manganese Bronze and Brass Co., Ltd., Birkenhead; and, altogether, this fine quadruple-screw ship represents a valuable addition to the tonnage available in this important service. Under her new name, the vessel has a service speed of about 18 knots, as compared with her original speed of 21 knots, economy being of prime importance.

Reference to turbo-electric machinery recalls that the Société Générale de Constructions Electriques et Mecaniques Alsthom, of Belfort—the builders of the turbo-electric machinery of the *Normandie*—completed a very fine twin-screw turbo-electric installation last year for the 24,000-S.H.P. trans-Mediterranean liner *Kairouan.* This vessel is designed for a service speed of 23 knots, current being generated by a pair of two-pole turbo-alternators which deliver 9020 kw. when driven at 3375 rpm. The initial steam conditions are 500 lb. per sq. in. pressure and 750 deg. F. temperature.

For economical running, one turbo-alternator is capable of supplying both propelling motors, so that a shaft output of 10,900 hp. at about 145 rpm. is available.

The pattern adopted successfully in the *Normandie* has, to some extent, been followed in the new ship, an interesting feature of which is that the main and auxiliary machinery is arranged in what is virtually four machinery spaces. The aftermost of these accommodates the two propelling motors. Forward of this is a boiler-room, which houses four La Mont forced-circulation boilers, designed for a steam pressure of 575 lb. per sq. in. and a temperature of 800 deg. F. at the superheater outlets; three of these boilers are sufficient to maintain full power.

An unusual feature is that the boilers have separate uptakes which do not unite until the boat-deck level, with appreciable improvement in the layout of the public rooms. The boiler circulating pumps are motor-driven units, and these, with the forced-draught fans and other auxiliaries, are installed in the boiler-room. The propulsion-control desk is in the next machinery space, which houses the two main turbo-alternators, distillers and various auxiliary pumps, although the driving motors of the main circulating pumps are at the other side of the bulkhead, i.e., in the forward machinery compartment. This contains two 1850-kw. auxiliary geared turbo-alternators and three 250-kw. Diesel alternators, which supply current for various purposes. The *Kairouan* was the only important European-built turbo-electric ship of last year, and it will be interesting to see in due course how she compares with geared-turbine vessels in the same service.

Before concluding this survey, it should be recorded that specialists in oil-burning appliances for marine purposes have of late given close attention to the improvement of their equipment. The quality of boiler oil available today is such that efficient combustion is not so easily obtained as it was at one time, while the march of progress in connection with boilers and machinery has made it desirable that the performance

of oil-burning equipment should reflect that advance. "Doping" of fuel oil is frequently resorted to in order to facilitate good combustion and moderate maintenance outlay.

One of the leading firms in the marine oil-burning field, the Wallsend Slipway & Engineering Co., Ltd., have recently introduced a new single-control air-distributing unit in connection with their widely-used Wallsend-Howden oil-burning system, so that all classes of lower-grade fuel oils can be burnt effectively and economically. With this improved equipment, atomization is very efficient and the formation of deposits on the heating surfaces is stated to be virtually eliminated, while (most important in passenger vessels) such deposits as do leave the funnel top are dry and free from oily or combustible matter which might settle on decks, boat gear, etc.

The vanes of the air deflector in this improved distributor impart a high-speed whirling motion to the combustion air, the whole device being so arranged that a short flame is obtained. All internal parts are fixed, so that no regulation or adjustment is required by the boiler-room staff. The burner is encircled by a tube along which passes an air supply, the burner being thereby kept cool and the formation of carbon deposits (in maneuvering for example) minimized.

At the same time, the firm have introduced an effective electric lighting-up torch for marine use, the current for this being provided by a battery which can at all times be maintained fully charged by means of a trickle-charging device operating from the main electrical circuit. There is no doubt that the elimination of paraffin, cotton waste and the like for lighting a burner is a desirable and practical step, and this equipment should prove no less popular than the improved oil-burning gear here briefly described.

COOLING OF ELECTRONIC APPARATUS AT HIGH ALTITUDES

ACKNOWLEDGMENT

This article was published in the "CADO" (Central Air Documents Office—Army, Navy, Air Force) Technical Data Digest," December, 1950. It was written by HARRY WILKIE, Engineering Division, Directorate of Research and Development, Headquarters, Air Materiel Command. The article outlines the advantages and disadvantages of several systems for cooling electronics apparatus at altitudes above 50,000 feet.

The problem of maintaining the operating temperature of airborne electronic apparatus within allowable limits has become increasingly difficult with increased operating altitudes of aircraft. Design trends point toward imposition of higher altitude conditions on the apparatus than has been considered heretofore. The marginal limits of conventional air cooling systems are now being encountered in propeller-type airplanes. Such systems may be expected to be unacceptable in future aircraft which will include high-altitude, jet-propelled designs.

Altitude tests on radiators for water-cooled engines have shown that the cooling effect of the atmosphere is proportional to the mass flow of air through the radiator. As the density of the atmosphere decreases with altitude, a progressively greater volume of air must pass through the radiator in order to maintain the same cooling effect. Considering altitude variation of atmospheric density alone, maintenance of the same cooling effect as at ground level would require a two-fold increase in volume of cooling air at 22,000 ft.; fourfold at 40,000 ft.; tenfold at 59,000 ft.; and hundredfold at 120,000 ft. These figures are based on the standard atmosphere used by the National Advisory Committee for Aeronautics.

Other tests on aircraft engine radiators have shown that for the same mass flow of air, the amount of heat dissipated is proportional to the temperature difference between it and the radiator liquid. Within certain lower altitude limits, atmospheric temperature decreases with altitude.

This improves the atmosphere as a cooling medium by both increasing the temperature difference and retarding the loss in density. The NACA standard atmosphere follows a series of straight lines connecting the following points: The ground-level temperature is 59° F., which decreases to —67° F. at 35,332 ft., remains constant to 104,987 ft., then increases to 170.6° F. at 164,042 ft., remains constant to 196,850 ft., decreases to —27.4° F. at 255,910 ft., remains constant to 272,310 ft., then increases to 215.6° F. at 393,700 ft. and further with increased altitude. This last altitude is well below those being reached in some rocket tests.

Here, the atmosphere would heat rather than cool many of our currently acceptable radio-circuit components. It is recognized that the standard atmosphere may not agree with actual conditions. However, it lies within the limits of error of available experimental data.

Combining and plotting the temperature-altitude and density-altitude data gives the curve shown in Fig. 1. The sharp knee defines the point beyond which ambient air cooling offers next to nothing. Somewhere before this point is passed, an altitude will be reached where the blower motor will not move enough air to dissipate the losses of it alone. The increase in volume of moved air that results from increased speed of the blower due to the lighter atmosphere was not included in Fig. 1.

Packing the apparatus with dry ice offers a simple solution of the cooling problems. However, the dry ice is not always easily obtainable and it has a low heat absorption per pound of material. It would require evaporation of 11.7 lb. of dry ice to absorb one kwhr. On the other hand, there may be some special application where it will prove to be the basis for a practical system.

Another proposed method allows the evolved heat to boil some material and subsequently discharge the heat overboard in the vapor. Under conditions where the apparatus would need insulation from other heat sources as in some jet-propelled aircraft, such a system seems appropriate. For a practical system it would be required that the expendable evaporant:

1. Have high latent heat of vaporization in order to limit the overall weight of the system, especially on long-time flights.
2. Be cheap.
3. Be easily obtainable.
4. Boil at a suitable temperature.

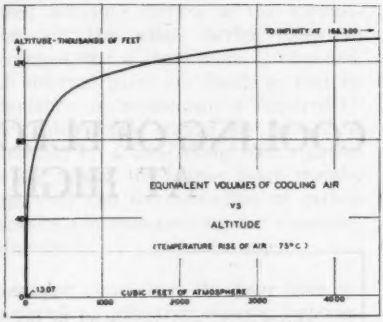


Figure 1

Figure 2 shows the boiling temperature versus altitude of a number of possible materials. These materials were selected because of wide distribution and ease of storage. Figure 2 also indicates the amount of heat absorbed by each in heating from 25° C. up to a boiling temperature of 45° C. plus the latent heat of vaporization. Methyl alcohol satisfies all requirements. It has a high heat of vaporization compared to most other materials. However, it is far exceeded in this respect by water. Heating and vaporizing 3.3 lb. of water is sufficient to dissipate a kilowatthour. The objectional feature of water is the high boiling point. This objection would be

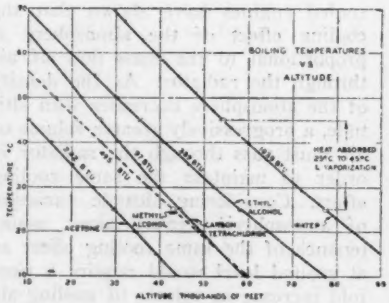


Figure 2

eliminated in the upper atmosphere. At 30,000 ft. water boils at 69.2° C. The material could be made satisfactory from about that point upward. An expendable water-evaporant system would prove satisfactory under conditions where the aircraft climbs to the stratosphere before the temperature of the total volume of liquid becomes too warm to protect the equipment. The liquid temperature could be limited for an extended period by initially adding the water as an ice and water mixture or the boiling temperature could be limited for a time by using a water solution of methyl alcohol.

For comparative purposes, two complete cooling systems were fabricated. Both are designed to remove the heat generated by electronic apparatus in a pressurized case designed for operating at high altitude.

In order to get an indication as to what degree of the required increased volume of cooling air might be provided by the increased speed of blowers driven by direct-current series motors, a number of blowers were tested in an altitude chamber. The results of these tests indicated that doubling the speed is about as much as can be expected at 70,000 ft. Assuming that there is no loss in efficiency of the blower fan, this would about double the volume of air moved at that altitude. For cooling effect equivalent to ground level, Fig. 1 indicates a volume increase of eight times is needed at that altitude.

It is apparent that air blower systems will be of no utility above an altitude which is still some distance below the 100,000-ft. knee on the curve of Fig. 1. On the other hand, it is reasonable to assume that where the atmosphere is sufficiently dense to support conventional propeller airplanes, it will be sufficiently dense to be utilized as a cooling medium. The official altitude record for propeller-type airplanes currently is 59,445 ft. Above this altitude, a new type of cooling is desirable and sometime before the knee

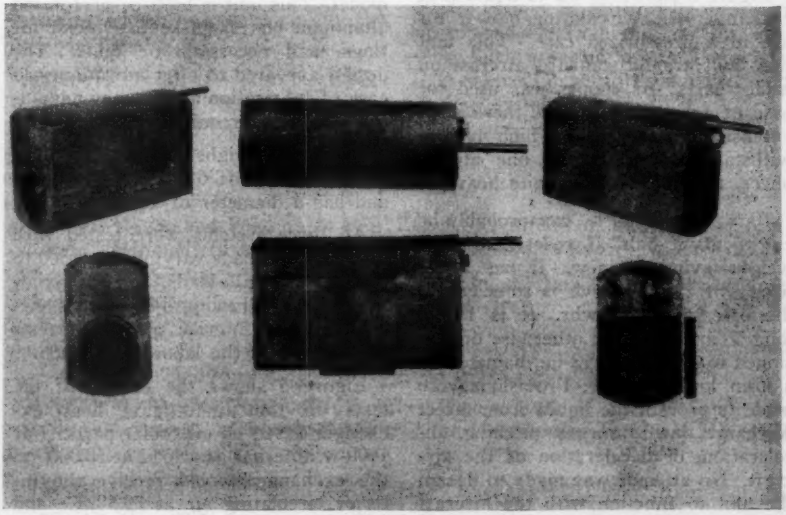


Figure 3

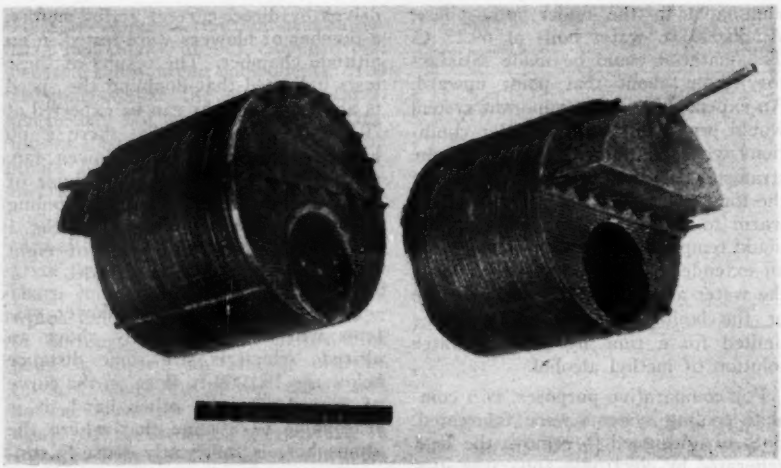


Figure 4

of Fig. 1 is reached, a new type, not dependent on the atmosphere, is necessary.

Figure 3 shows an air-to-liquid heat exchanger with circulating blower for use in a pressurized case. This unit was built around standard aircraft oil cooler parts. As copper was used for ease of fabrication and as there were extensive flat surfaces which had to withstand a pressure of one atmosphere, the unit became quite heavy.

Its weight of 50 lb. can probably be cut to about half, if redesigned with weight-saving in view. It has a capacity of 2 gal. which is sufficient to dissipate about 5 kwhr. It is 19 in. long, 7 in. wide, and otherwise dimensioned to slip into a 12-in. diameter by a 36-in. long test case. Provisions were made to prevent the liquid from direct discharge due to surges, pitching, acceleration, or deceleration of the aircraft. No attempt was made to design the unit to function with the aircraft continuously inverted.

Figure 4 shows a second air-to-liquid heat exchanger with circulating blower for use in the same pressurized test case as was used with the previous model. This unit is built of perforated aluminum plates, gaskets, two-end fittings, and necessary tie bolts. The design appeared to offer advantages in ease of fabrication and in easy adaptation to varied requirements.

The unit weighs 33 lb. It has a liquid capacity of 6 qt., is 14 in. long, and has a diameter of 11⅞ in.

All three systems were tested with internal heat loads varying from 500 to 1000 w. At all loads, a 5-in. axial-flow blower drawing 140 w. was circulating the interior air. The tests were made on the laboratory bench.

Figure 5 shows the results of the tests on the air-to-liquid heat exchanger based on oil cooler parts. For 1000-w. internal heat load at 50,000 ft., the exchanger would receive the interior circulating air at 79.5° C., and return it at 72° C. These temperatures

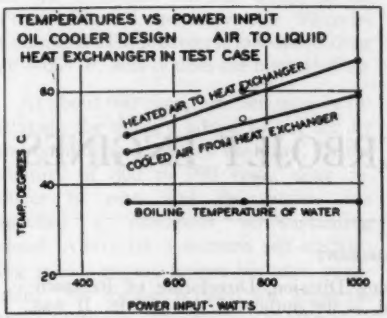


Figure 5

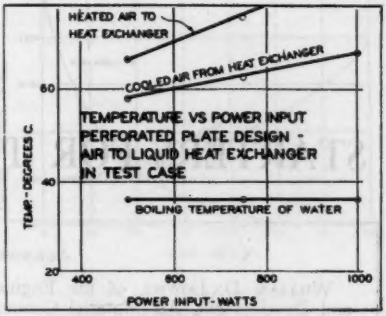


Figure 6

would be 61.5° C. and 54° C., respectively, at 70.000 ft. This represents a temperature rise of 44° C., over the temperature of the boiling water.

Figure 6 shows the results of the tests on the air-to-liquid heat exchanger designed around the perforated aluminum plates. To duplicate the temperatures obtained on the previous air-to-liquid exchanger would require reducing the heat load to 500 w. The same interior blower was used as in the previous case.

Both these air-to-liquid heat exchangers had about the same area exposed to the interior circulating air. The air velocity in the perforated plate design was a little more than half of that in the other. This, plus the longer heat path in the metal, probably accounts for the inferior performance of this model.

It is believed that these tests have demonstrated that the cooling needs of electronic apparatus at altitudes above 50,000 ft. may be satisfied by means of an expendable evaporant cooling system. Such a system offers advantages over conventional or known proposed cooling systems.

References

1. James, W. S., and Parsons, S. W., "Effect of Altitude on Radiator Performance," *National Advisory Committee for Aeronautics* Report No. 62. 1920.
2. Diehl, W. S., "Standard Atmosphere—Tables & Data," *National Advisory Committee for Aeronautics* Report No. 218. 1925.
3. Warfield, C. N., "Tentative Tables for the Upper Atmosphere," *National Advisory Committee for Aeronautics* Technical Note No. 1200. 1947.
4. Wood, G. P., "Calculation of Surface Temperature in Steady Supersonic Flight," *National Advisory Committee for Aeronautics* Technical Note No. 1114.
5. Williams, Jr., "A. D. Little's New Heat Exchanger," *Chem. Eng.* Vol. 56, pp 104-107. 1949.

STARTERS FOR TURBOJET ENGINES

ACKNOWLEDGMENT

WILLIAM D. DOWNS, of the Engineering Division, Directorate of Research and Development, Air Materiel Command, is the author of this article. It was published in "CADO (Central Air Documents Office, Army, Navy, Air Force) Technical Data Digest" for February, 1951. The author discusses development of air starters and cartridge starters for turbo jet engines.

INTRODUCTION

Present trends indicate that starters rated at more than 1000 hp. will be used on turbojet engines in the near future. Less than twenty years ago 1000 hp. was the objective of aircraft engine development.

Our largest reciprocating engine, the 3500-hp. R4360 engine, requires only a 5-hp. starter. The reason for this disparity of starting-power requirements is this: a reciprocating engine requires starting assistance to only 2 or 3% of its rated speed, whereupon ignition may be initiated and the engine will develop sufficient power for satisfactory acceleration; on the other hand, turbine engines require assistance of a starter to 20% or more of rated speed. Since power is a function of speed and torque, the high-speed cranking is the reason for the higher starting power required by gas turbines.

STARTING DATA FOR JET ENGINES

The drag and torque data of a typical turbojet engine, Fig. 1, will illustrate the starting characteristic of a gas-turbine engine. At rest, a breakaway drag is encountered. The breakaway drag has been omitted from this chart. On engines tested by the writer, breakaway drag has been smaller than the peak running drag. At low speed the engine drag is low. Engine drag increases as speed increases in a fashion typical of a blower. The values of drag shown on Fig. 1 are for normal temperatures. At —65° F., the increased density of charge will increase the drag by about 35%. A much greater increase may be caused by mechanical friction.

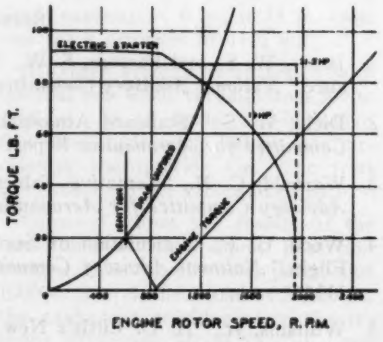


Figure 1

In one axial-flow turbine tested at the Power Plant Laboratory, Air Materiel Command Headquarters, low-speed drag at —65° F. was double the normal drag.

At about 600 rpm., ignition may be initiated; the energy added to the air by the combustion reduces the drag. In the vicinity of 800 to 900 rpm., drag reduces to zero and the engine has reached a minimum self-sustaining speed. Above the minimum self-sustaining speed, engine torque becomes available. This torque is used for acceleration. However, the torque available is not adequate for satisfactory acceleration until the engine reaches the starter cut-out speed, 1600 rpm., or slightly in excess of 20% of maximum rotor speed, which is 7950 rpm. Rated thrust of this engine is 5000 lb. This engine uses an electrical starter. The torque characteristic of the electrical starter is approximately as shown. Current and torque are held very nearly constant from stall to 900 rpm. engine rotor speed by varying the voltage to the starter. The torque of an electrical starter using a constant voltage source would normally peak at stall and decrease with increasing speed. Assuming ignition at 600 rpm., this starter will accelerate the engine to 1950 rpm. in approximately 36 sec. The peak power developed by this starter is 19 hp.

A constant-torque device which delivers 85 lb. ft. of torque, 31.5 hp. at 1950 rpm., is required to provide the same duration of start. This brings out the fact that power rating of a starter is not fully descriptive of the capabilities of a starter. The shape of the curve of starter torque versus speed must also be considered in predicting the performance of the starter on a given engine. Therefore, the power of a constant-torque starter which gives performance equivalent to the performance of a given starter will be referred to as effective horsepower of that starter.

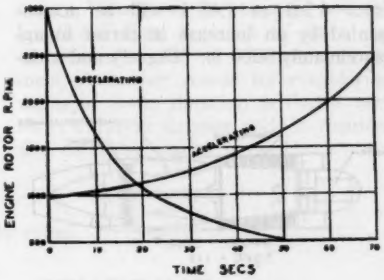


Figure 2

The starting data which have been considered were determined from deceleration runs and acceleration runs on the engine. The formula used to derive the data is as follows:

$$T = \frac{P}{32.2} \times \frac{6.28}{60} \frac{dN}{dt}$$

where T is torque, lb. ft.

P is polar moment of inertia, lb. ft.^{2}

N is engine rotor speed in rpm.

t is time in seconds

Figure 2 shows the data of a deceleration run and an acceleration run. The data of Fig. 1 for drag and engine torque were calculated from this information. During the deceleration run, fuel was cut off so that there was no combustion. During the acceleration run the gas temperature in the tail pipe was maintained at a constant value of 1275° F. It may be noted that the time to accelerate from 2000 rpm. to 3000 rpm. may be reduced by approximately 50% by raising the tail-pipe temperature 200° F.

The reciprocating engine went through the torments of being supercharged not long after it became airworthy. If a similar fate overtakes our turbojet engine which is rated at 5000 lb. thrust, an increase in pressure ratio

from 4.5:1 to 7.45:1 will be accompanied by an increase in thrust to approximately 9200 lb. (Fig. 3) and a de-

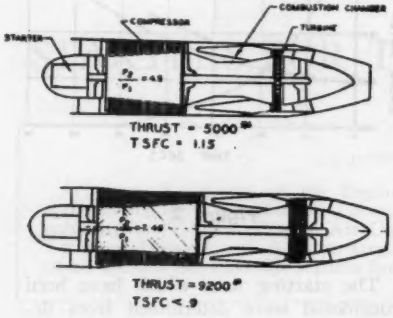


Figure 3

crease in fuel consumption from 1.15 lb./hr./lb. of thrust to less than 0.9 lb./hr./lb. of thrust. The redesign will be accompanied by an increase of compressor temperature rise from 325° F. to 480° F. at design speed and 60° F. inlet air temperature. This increase in temperature rise will necessitate building a greater portion of the compressor using steel rather than aluminum. This, together with the increase of compressor and turbine stages, will increase the moment of inertia of the engine rotor. The increase in engine thrust was brought about by an increase in pressure ratio and an increase in air flow through the engine. At design speed the air flow is increased by approximately two-thirds. Increased air flow and pressure rise will increase the engine rotor drag during starting. Another effect of the redesign is a more rapid deterioration of compressor efficiency as speed is decreased. Evidence of this effect is an increase in engine idling speed. Whereas the engine which delivered 5000 lb. of thrust idled at about 27% of rated speed, engine manufacturers have estimated that an engine such as the 9200-lb.-thrust engine may have an idling speed as high as 50% of rated engine speed. Hence, a considerable increase in the starter-disengagement speed must be expected.

The increase in engine-rotor moment of inertia, the increase of engine drag, and the increase of starter-disengagement speed will increase the power required to start the high pressure-ratio engine. Figure 4 presents curves of

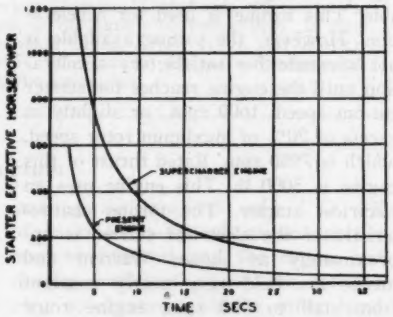


Figure 4

estimated torque required versus starting time for the 5000-lb.-thrust engine and for the 9200-lb.-thrust engine. The curves show that the 5000-lb.-thrust engine will require approximately 60 hp. for a 20-sec. start and approximately 400 hp. for a 3-sec. start; the 9200-lb.-thrust engine will require approximately 180 hp. for a 20-sec. start and 1200 hp. for a 3-sec. start. More than a sixfold increase of starter power is required to reduce the starting cycle from 20 sec. to 3 sec. Consequently, high-strength drives will be required for engines which are to be started in a 3-sec. duration. Such high-strength engine drives may in some instances lead to increased engine weight. However, when one considers the interceptor aircraft for which seconds may mean the difference between a successful and an unsuccessful interception mission, one

may then argue that the time saved by the fast start is equivalent to a drastic increase in engine thrust in terms of the length of time required to effect the interception. This argument may be countered by questions concerning the length of time required for a pilot to secure himself in the aircraft and the length of time required to warm up essential electrical or electronic equipment. A further reason for considering starts of 3-sec. duration is that a very short duration starting cycle is required if a cartridge starter is used.

STARTER TYPES

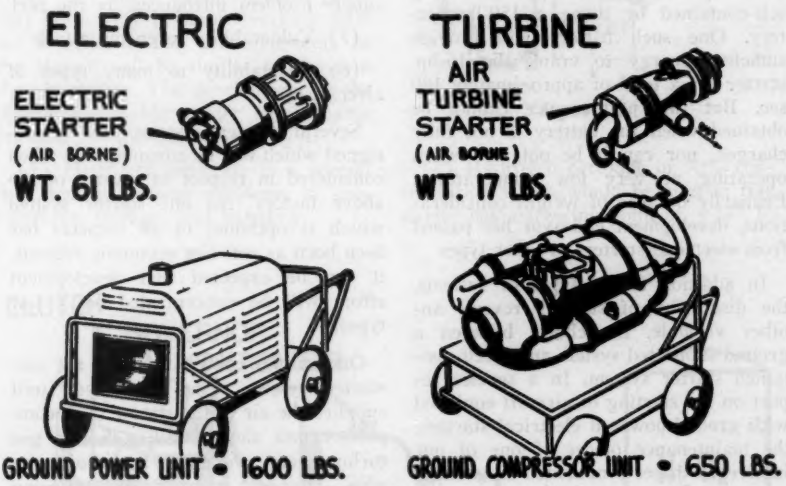


Figure 5

Aircraft presently in service are started either by using electrical starters or starter generator. The starter generator is very advantageous from the point of view that the starter will serve as a generator while the aircraft is in flight. Therefore, most of the weight of the motor may be charged to the generator duty. However, reliability of a generator which is subjected to the strains of starting remains doubtful. Figure 5 shows the starter elements of an electrical system and an air-starter system. The 61-lb. electrical starter is rated at 16 hp. The 17-lb. air starter is rated at 35 hp. The air starter will provide a faster acceleration for the turbojet engine with a weight saving of 44 lb. per engine. In addition, large weight savings in ground equipment can be anticipated. The generating set for the 16-hp. electrical starter weighs 1600 lb. The gas turbine which provides the air for the 35-hp. air starter weighs only 88 lbs. Assembled on a trailer with fuel cells and accessories, it is expected to provide a transportable ground power source which will weigh not more than 650 lbs.

The systems as illustrated in Fig. 5 are referred to as ground-supported

starter systems since the prime mover remains on the ground. An aircraft is said to have a self-contained starter system if the fuel and the device for converting the chemical energy of the fuel are contained in the aircraft.

The air starter is readily adaptable as a self-contained starter system. It is lightweight and it will use the fuel of the aircraft. The 16-hp. electrical system may be made to approach being self-contained by using a 150-lb. battery. One such battery will provide sufficient energy to crank the 16-hp. starter for a total of approximately 180 sec. But this performance cannot be obtained when the battery is not fully charged, nor can it be obtained when operating at very low temperatures. Primarily because of weight considerations, development endeavor has passed from electrical starters to other types.

In addition to the types of systems, the discussion of starters reveals another variable, the choice between a ground-supported system and a self-contained starter system. In a service report on the starting of aircraft equipped with ground-powered electrical starters, the maintenance officer of one of our interceptor bases commented on ground-supported systems as follows:

> "Even if the proper amount of power plants, operating personnel, vehicles, and drivers were available, the cost of equipment and the expenditure of manhours for starting a large number of aircraft in a short time is excessive beyond a doubt."

Tactical considerations will require that interceptors and some other types of military aircraft be equipped with self-contained starter systems.

Three factors requiring consideration in the choice of a starter system have thus presented themselves:

(1) Weight.

(2) Suitability for operation under extreme cold temperatures.

(3) Suitability as a completely airborne system.

Other factors which must be considered are:

(4) Bulk.

(5) The number of starts which may be made without refueling.

(6) If a special fuel is needed, the cost, fabrication, storage, handling, and supply problem introduced by the fuel.

(7) Vulnerability to enemy attack.

(8) Adaptability to many types of aircraft.

Several starter systems can be designed which will be advantageous when considered in respect to several of the above factors. No one starter system which is optimum in all respects has been born as yet. For economic reasons, it can be expected that development effort will be concentrated on a few types.

One of the chosen types is the air-starter system. The compressor unit supplies the air to the starter. The compressor unit shown in Fig. 5 is a gas turbine very similar to a turbojet engine. However, instead of supplying available energy as a propulsion jet, this gas turbine supplies available energy as compressed air.

Figure 6 is a schematic drawing of an air turbine starter. The available energy of the compressed air is extracted by a high-speed turbine wheel. The reduction gear reduces the speed to a suitable cranking speed for the engine. A speed sensing device indicated as a governor in the sketch will cause the air to the starter to be shut off when maximum cranking speed has been reached. The mechanism indicated as a clutch is an automatic engaging and disengaging mechanism. This mechanism is required because the starter turbine is designed to reach its

peak allowable speed when the turbojet is at probably 20 to 35% of its operating speed.

Figure 7 shows a sketch of the hydrogen-peroxide turbine starter. Hydrogen peroxide is fed by a pump to a catalyst chamber. The catalyst chamber is operated at 300 psi. The catalyst decomposes the hydrogen peroxide into steam and oxygen at approximately 1350° F. These decomposition products are a very desirable working fluid for a turbine. The drive elements of the starter will be similar to those in the air starter. The development of the hydrogen-peroxide starter is not being pursued for the following reasons:

(1) Hydrogen peroxide is a special fuel item presenting problems in procurement, storage, and handling.

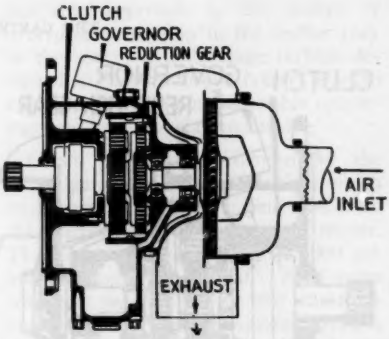


Figure 6

(2) Hydrogen peroxide of 90% concentration which would be used in this starter has a freezing point of +13° F.

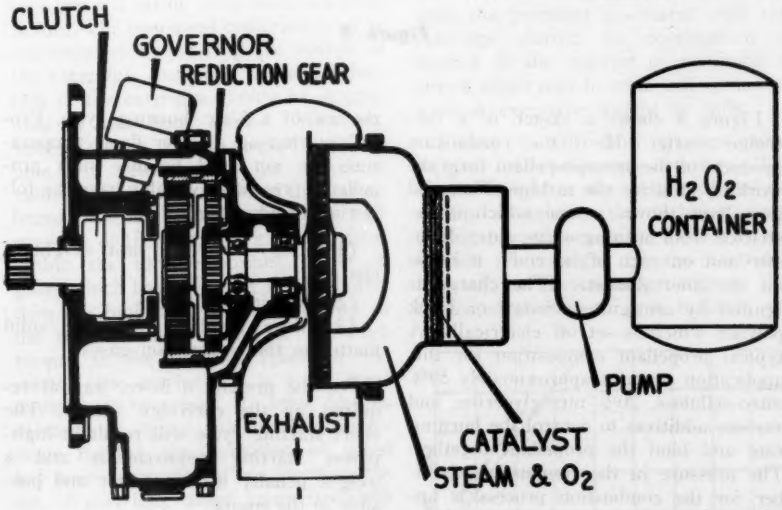


Figure 7

THE CARTRIDGE STARTER

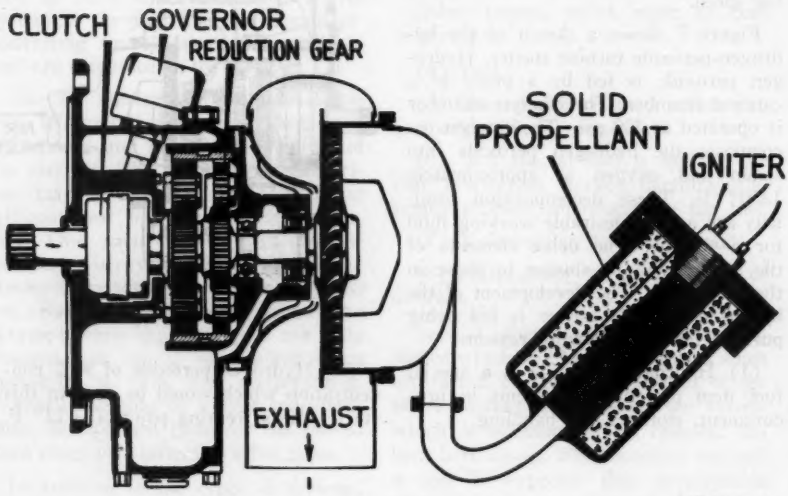


Figure 8

Figure 8 shows a sketch of a cartridge starter. Here the combustion products of the monopropellant form the working fluid for the turbine. The solid propellant shown is one which is restricted from burning on its outer diameter and on each of its ends; it burns on its inner diameter. The charge is ignited by an igniter charge of black powder which is set off electrically. A typical propellant composition for this application contains approximately 59% nitro-cellulose, 20% nitroglycerine, and various additives to control the burning rate and bind the propellant together. The pressure in the combustion chamber for the combustion process is approximately 1000 to 1500 psi. The flame temperature of the combustion products is approximately 3000° F. The high flame temperature leads to the requirement that the starter operates with a burning cycle of very short duration. Maximum success has been obtained by the use of a 3-sec. burning cycle. Propellants having a lower flame temperature are not used because such propellants presently available have the following disadvantages:

(1) Less energy per unit weight of charge.

(2) Ignition difficulties.

(3) More residue and more solid matter in the combustion gases.

For the present, a 3-sec. start is required by the cartridge starter. The short starting cycle will result in high-power starting requirements and a weight penalty in the starter and possibly in the engine.

The solid propellants presently available are temperature sensitive. That is, the burning rate varies with the temperature of the propellant. The variation is approximately 0.2% per degree Fahrenheit. Hence, starter power falls

at low temperature, whereas the power required by the turbojet increases.

There is also a storage problem. Propellants presently available tend to crack when stored below —40° F. Cracking increases the burning area and the increase is likely to be so great that an explosion may occur if the propellant is ignited.

The propellant is an extra supply item; however, it is similar to the propellants used in solid rocket ammunition and in solid-propellant JATO devices.

In addition to the flame temperature problem there are design problems in the cartridge starter involving safety devices. These concern protection against explosion, protection against firing when the breech is not fully secured and protection against overspeed. The overspeed protection is a most serious problem. Overspeeding will result in the event of a drive failure. The overspeed condition must be counteracted by wasting the energy of the cartridge, inasmuch as the combustion of the cartridge cannot be stopped once it has been started. Moreover, starter torque for a single-stage turbine will remain near its maximum value at the limiting speed of the starter, because the high temperature and high pressure of the cartridge gases produce, within the turbine nozzles, a jet of gases which has a velocity much greater than the limiting pitch line velocity of the turbine. As a result of the high torque at the limiting speed of the starter, the starter has a very great tendency to overspeed. In some designs of cartridge starters employing a single-stage turbine, the limitation of speed has been accomplished by diverting the jet of cartridge gases away from the turbine wheel or by removing the turbine wheel from the jet of cartridge gases. Because of the great tendency to overspeed, a negligible time delay is required of any device chosen to sense the overspeed and move parts to counteract the overspeed condition. A more desirable approach to the design of overspeed protection in the starter may be the use of a multi-stage turbine designed so that its torque falls to a small value at the maximum allowable operating speed of the starter turbine.

Other problems introduced by the cartridge starter involve the duct which must pass through the engine inlet from the cartridge housing to the starter. This duct will carry gases at 1000 psi. and approximately 3000° F. Engine problems introduced by the cartridge starter involve the fuel system, ignition system, and combustion chamber of the engine. These must be such that the engine can light off satisfactorily within the 3-sec. period of time and at the high speed to which the starter will accelerate the engine; otherwise the energy spent by the starter will be wasted when the engine slows down to a speed at which it can satisfactorily start. Despite the problems associated with the cartridge starter, its development is desired in the interest of obtaining a device which may be made self-contained without excessive weight or bulk.

Figure 9 will show the estimated weights of turbine-type starters. The starter weight is the weight of the assembly containing the turbines, gear, clutch, and drive units. This will not include the weight of the cartridge housing for the cartridge starter. The estimated weights of gas turbines are

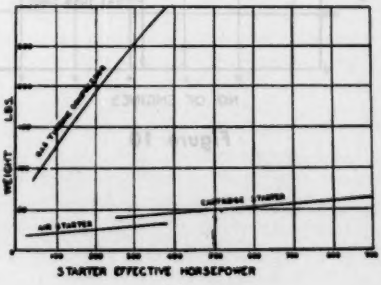


Figure 9

plotted against a horizontal scale of power of the starter for which they can supply compressed air. From the starter torque estimate for the 9200 lb. thrust engine, it is found that the starter power required for a 20-sec. start is 180 hp. and the starter power required for a 3-sec. start is 1200 hp. However, for the 20-sec. start an air starter having peak torque at stall with torque falling to zero at 125% of disengaging speed will be used. This will reduce the peak power of the air starter to 130 hp. for an effective 180 hp. The weight of the cartridge starter is 78 lbs.; the weight of the air starter is 28 lbs.; and the weight of the gas turbine which supplies compressed air to the air turbine is 175 lbs.

Figure 10 shows a comparison of the weight of the air starter and of the cartridge starter in single-engine and multi-engine aircraft. Tubing, connecting the nacelles of a multiengine aircraft, is not charged against the air starter inasmuch as that tubing will normally be utilized for carrying air used in cabin charging, deicing, and pneumatic-accessory drives. For the air-starter system, only one 175-lb. gas-turbine compressor is required regardless of the number of engines. To account for the installation equipment required by the gas-turbine compressor, 65 lbs. were added to the air-starter system. The cartridge was estimated to weigh 10 lbs.; the housing for the cartridge weighs 25 lbs. The cartridge housing is empty but two cartridges are stowed in the aircraft. The single-engine installation of the cartridge starter weighs 128 lbs. A six-engine installation of the air starter weighs 453 lbs.

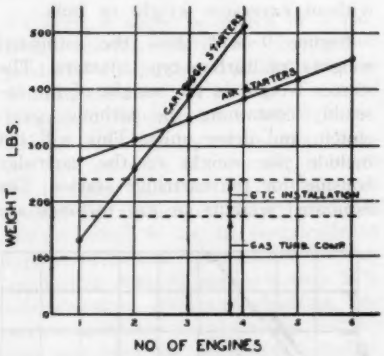


Figure 10

Both from the viewpoint of weight and volume, the cartridge starter will be preferred over the completely self-contained air starter for single-engine and twin-engine aircraft. However, if the gas-turbine compressor unit is installed in a detachable blister or pod, the compressor unit then may be carried, making the aircraft self-sufficient while moving from one air field to another; or, on the other hand, the compressor unit and its blister could be detached for combat missions. Then only the 28-lb. starter and the weight of some tubing could be charged to an air starter. The air-starter system may find use in some single- and twin-engine jet-propelled aircraft.

The cartridge starter may find application on multiengine aircraft in an installation such that one or two engines are started by the cartridge starter. The remaining engines would be equipped with air starters which would be energized by air bled from the first engine started.

References

1. Gregory, A. T., and Pomeroy, A. L., "Future Trends in Aircraft Engine Design," paper presented before SAE national Aeronautical Meeting. 9-11 April 1947.
2. Bardwell, A. G., Jr., "Aircraft Turbojet and Gas Turbine Engine Starter System," "Mechanical Engineer" June 1947.
3. Wood, H. J., and Dallenbach, J., "Auxiliary Gas Turbines for Pneumatic Power in Aircraft Applications," paper presented before SAE National Aeronautical Meeting. 5-8 October 1949.

WELDING PROCEDURES FOR HIGH-PRESSURE, HIGH-TEMPERATURE STEAM PIPING

ACKNOWLEDGMENT

This article was published in The Welding Research Supplement of "The Welding Journal," January, 1951. Its author, N. L. NAVARRE, Welding Engineer, U. S. Naval Engineering Experiment Station, presented it at the Thirty-first Annual Meeting of the American Welding Society, in Chicago, during the week of October 22, 1950. The paper discusses the development of welding procedures for alloys intended for high-temperature and high-pressure service, including preheat and postheat treatments for various alloy combinations, electrode selection and thermal shock fatigue tests.

INTRODUCTION

The evolution of the present high-pressure, high-temperature steam system was accelerated tremendously during the past two decades. In the early thirties, temperature and pressure were approximately 440°F. and 400 psi., respectively. By the late thirties, most of the capital ships of our Navy had been modernized and were operating under 650° F. and 900 psi. steam conditions. Today, steam-generating systems operating at 1050° F. and 2000 psi. are under construction. Thus, over the span of a relatively few years we have seen reciprocating engines replaced by high-speed turbines and flanged connections replaced by welded joints. To satisfy the demand for steam generation at increased pressures and temperatures, new alloys had to be developed. For 650° F. steam service, carbon-molybdenum alloy steel was produced. The Springdale failure in 1941 resulted in the addition of chromium to carbon-molybdenum alloy steel for stabilization against graphitization at prolonged elevated temperature service. The demand for greater strength in the present-day steam systems is being met by still higher percentages of the alloying elements such as chromium and molybdenum, and in some cases by the use of austenitic steel.

The great strides taken in steam power systems could hardly have been possible without the scientific advances made in welding. Each new alloy introduced new problems which had to be solved before the fabricator could install the system. First, the electrode producers had to furnish electrodes that would yield weld metal with mechanical properties comparable to those of the base metal under the intended service conditions. Secondly, welding procedures had to be developed that would not impair the original properties of the base material. The mere fact that austenitic steel systems are

being considered for high-temperature, high-pressure steam service brings to mind at least one problem requiring immediate solution to permit effective use of the applicable materials. This problem is the welding of joints comprised of materials having considerable difference in expansion coefficients, i.e., ferritic vs. austenitic joints.

The Navy has pioneered in the application of carbon-molybdenum steel piping and turbines for the 650° F. installations, which started in the late thirties. It is now pioneering in the application of new materials for steam generation at pressures up to 2000 psi. and temperatures up to 1050° F. The role of the Welding and Electrical Laboratory of the Experiment Station is to perform investigations and tests to develop welding procedures for new materials. This paper presents the development of welding procedures intended for the above given high-pressure, high-temperature steam service.

Pre- and Postheat Treatment Determination

The base materials employed in this investigation were as follows:

(*a*) ½% chromium
½% molybdenum
(*b*) 1% chromium
½% molybdenum
(*c*) 2¼% chromium
1% molybdenum
(*d*) 4-6% chromium
½% molybdenum
(*e*) 18% chromium
8% nickel (columbium)

Each of the above materials was submitted to a synthetic weldability test† developed at the Experiment Station. In brief, this test consists of determining the relative notch toughness of the steels prior and subsequent to the application of a synthetic thermocycle of welding. Notch toughness is defined as the radius about which a ⅜- × 1½-in. bar, containing a specially designed notch at midlength, bends 180° without failure. The radius is given as a multiple of the specimen thickness. The synthetic thermocycle of welding is applied by heating a ½-in. wide band, at midlength of the specimen, to 2150° F. from room temperature in 49 sec. by high-frequency induction. The metallurgical changes

Table 1—Preheat and Postheat as Determined by EES High-Frequency Notch-Bend Test

Material, %	*Preheat, ° F.*	*Postheat, ° F.*
½ Cr-½ Mo	300-400	1275-1300
1 Cr-½ Mo	400-500	1275-1300
2¼ Cr-1 Mo	400-600	1300-1325
4-6 Cr-1 Mo	500-600	1300-1325, Cool to 900 and hold
18 Cr-8 Ni	None*	None*

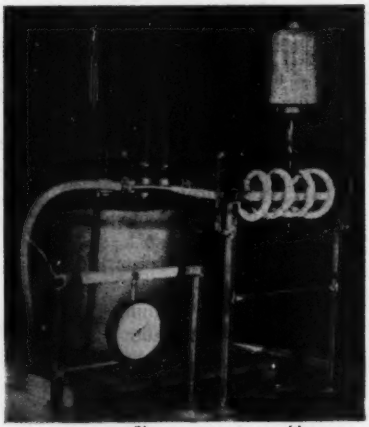
* Test not applied to this material. Data included merely to complete tabulation.

† "A Method for Determining the Applicable Welding Procedure for Steels," Ronay Bela, "The Welding Journal," 25 (11), Research Suppl., 736-s to 747-s (1946).

produced are comparable to those which occur in the heat-affected zones of multiple pass manual arc welds. In the event the notch toughness of the steel is impaired by the application of this synthetic thermocycle of welding, treatments at various degrees of preheat and/or postheat are applied until a combination is found that permits the base material to bend around the original radius. Thus, the applicable treatments for materials given in Table 1 were determined without actual welding.

STRESS RELIEF TEMPERATURE DETERMINATION

The performance of the high-frequency notched-bend test yields information which determines whether the metallurgical conditions in welded structures are satisfactory, particularly in regards to notch toughness. However, previous experience with the 4-6% chromium-½% molybdenum type steel showed that the temperature applicable to obtain stress relief may vary considerably from that required to restore notch toughness subsequent to welding. Therefore, the applicable stress relief treatment was determined for each of the ferritic alloys given in Table 1. The procedure employed, Fig. 1, is somewhat qualitative but it has proved to be effective for the purposes intended. It employs a rectangular beam measuring approximately 0.3 × 0.4 × 11 in. as the test specimen. This beam, Fig. 1 (*a*), is supported horizontally within a high-frequency induction coil and loaded at midlength to the maximum fiber stress of 3000 psi. Subsequently, the beam is heated by high-frequency induction, while under load, to a pre-determined temperature level which is selected arbitrarily. The holding time at this temperature is approximately 20 min. After cooling to room temperature, the amount of permanent set is measured employing a dial indicator of 1/10,000 in. sensitivity, Fig. 1(*b*). In the event no permanent set has occurred, the procedure is repeated at increasingly higher temperature levels until a permanent set of approximately 0.015 in. is obtained. The temperature which



(b) (a)

Fig. 1 EES loaded beam test for stress-relief temperature determination

(a) loaded beam in heating coil; (b) measurement of permanent set

TABLE 2—STRESS RELIEF TREATMENT AS DETERMINED BY EES LOADED-BEAM TEST

Material, %	*Stress Relief Treatment,* ° F.*
½ Cr-½ Mo	1275
1 Cr-½ Mo	1275
2¼ Cr-1 Mo	1300
4-6 Cr-½ Mo	1150
18 Cr-8 Ni	None†

* Holding time to be 2 hr. per inch of metal thickness.

† Stress-relief temperature determination test not applied to this material. Data included merely to complete tabulation.

produces this permanent set is then taken as the stress-relieving temperature of the metal, since at this temperature all stresses over 3000 psi. were relieved by plastic deformation. The stress-relief treatments obtained by the above method are given in Table 2. Examination of the data contained in this table revealed a stress-relief temperature of 1150° F. for 4 - 6% chromium - ½% molybdenum, while the postheat treatment given for the same material in Table 1 is 1300-1325° F. followed by a holding period at 900° F. This indicates that even though the former treatment is adequate to relieve residual stresses, it is not adequate to restore the original notch toughness of the material. Since it is obvious that the latter treatment will serve both purposes without changing the metallurgical structure of the material, it was selected as the proper postheat treatment for 4-6% chromium-½% molybdenum alloy steel.

RESIDUAL STRESS DETERMINATION

The findings of the induction-heated notched-bend test were substantiated by actual welding tests. In order to ascertain the validity of the stress-relief treatment, stress analysis was performed for a welded 10 in. IPS Schedule 160½% chromium-½% molybdenum alloy steel tube joint to which the applicable stress-relief treatment of Table 2 had been applied. The method of stress analysis employed for this purpose consisted of dissecting the assembly into 3-in. lengths and then relaxing each section by making a cut through one wall thickness. The resultant deflection was measured on tram points inserted along generator lines, in a plane perpendicular to that in which the saw cut is made. Subsequently, considering a quadrant of the tube as a curved beam fixed at one end and with a concentrated load at the other end, the stress is computed for a unit length of the tube section employing the following formula:

$$S = \frac{4EId}{R^2 M}$$

where S is the stress in psi., E is the modulus of elasticity, I is the moment of inertia of a unit length of the tube section, d is the deflection obtained for the quadrant of the tubing under consideration (change in diameter of tubing divided by two), R is the original mean radius of the tubing and M is the section modulus of a unit length of the tube section.

The results of the stress survey performed by the above described relaxation method for a welded 10 in. IPS ½% chromium-½% molybdenum alloy steel tube joint which was stress relieved at 1275° F. for 3 hr. are given on Fig. 2. Examination of these results indicate a negligible circumferential stress of 1750 psi. in the weld zone (center of stress-relieved zone). Accordingly, the stress-relief treatment of 1275-1300° F. is considered adequate to eliminate residual stresses. On the other hand, the unaffected portion of the tubing was found to have a circumferential stress as high as 12,000 psi. This condition indicates that the tubing as-received was not in the annealed, stress-free condition.

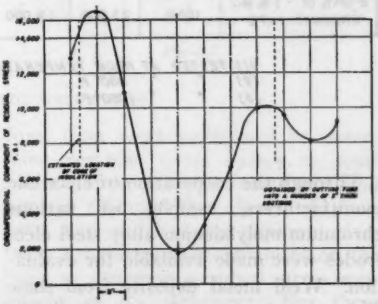


Fig. 2 Results of stress survey by relaxation method

Electrode Development

Table 3—Results of Weld Metal Tests

BASE METAL CLASSIFICATION	ELECTRODE CLASSIFICATION	TYPE OF COATING	MECHANICAL PROPERTIES			
			T.S (psi)	Y.P. (psi)	ELONG. (%) 2"	RED. IN AREA %
1/2% Cr. - 1/2% Mo.	1/2% Cr. - 1/2% Mo.	CELLULOSE	77,500 (1) TO 96,000 (1)	69,800	26.0	70.5
				69,800	24.0	67.6
			54,300 (3) TO 69,000 (3)	37,000	24.0	64.9
				52,500	16.5	57.8
1% Cr - 1/2% Mo.	1% Cr. - 1/2% Mo.	"	85,000 (1) TO 99,500 (1)	70,000	22.5	63.2
				85,500	22.0	63.8
			63,500 (3) TO 77,000 (3)	48,100	26.0	66.8
				65,500	21.5	60.6
2-1/4% Cr. - 1% Mo.	1-1/2% Cr. - 1/2% Mo.	LOW HYDROGEN	86,700 (1) TO 91,000 (1)	68,100	26.0	69.3
				78,000	23.0	59.5
			56,500 (3) TO 60,500 (3)	41,900	27.5	65.1
				47,500	30.0	76.5
	1-3/4% Cr. - 1% Mo.	"	88,200 (1) TO 96,000 (1)	71,500	24.5	70.8
				80,200	20.0	68.3
			55,500 (3) TO 61,800 (3)	42,400	24.8	64.8
				51,500	18.0	36.0
	2-1/4% Cr. - 1% Mo.	"	97,300 (1) TO 100,000 (1)	76,500	22.0	48.0
				81,700	22.0	66.8
			57,000 (3) TO 59,200 (3)	40,500	26.0	76.2
				43,200	21.0	74.6

ELECTRODE CLASSIFICATION	STRESS RUPTURE				CREEP		
	TEMP. °F.	STRESS (psi) TO CAUSE FRACTURE IN:			TEMP. °F.	STRESS (psi) FOR 1% MIN. CREEP IN:	
		100 Hrs.	500 Hrs.	1000 Hrs.		10,000 Hrs.	100,000 Hrs.
1-1/2% Cr. - 1/2% Mo.	1050	30,000	24,000	22,000	1050	10,000	5,000*
2-1/4% Cr. - 1% Mo.	1050	35,000*	24,000	20,000	1050	11,000	6,000
2-1/4% Cr. - 1% Mo. WROUGHT PIPE	1050	23,000	19,000	17,500	1050	12,000	6,500

(1) TESTED AT ROOM TEMPERATURE.
(2) " " 950° F.
(3) " " 1050° F

*EXTRAPOLATED

Through the cooperation of electrode manufacturers, samples of various chromium-molybdenum alloy steel electrodes were made available for evaluation. Weld metal deposits from these electrodes were analyzed chemically for screening purposes. Then the electrodes were submitted to a usability test consisting of deposition in the vertical by means of an automatic welding head, thus eliminating the human element from the welding oper-

ations. The latter test furnishes information relative to the current range of the electrode, the arc stability of the electrode within this range and the uniformity of the burn-off rate throughout the length of the electrode. Materials which passed the above tests successfully were put through the complete test procedure which yields information as follows:

(*a*) Soundness of the weld metal as determined by radiography.

(*b*) Static mechanical properties of the weld metal at room temperature and at the intended service temperature.

(*c*) Stress rupture and creep testing of the weld metal at the intended service temperature.

(*d*) Dilatometry of the weld metal as compared to that of the base metal.

A summary of data obtained for numerous chromium-molybdenum alloy steel electrodes tested as outlined above, is given in Table 3. Dilatometric curves for the various chromium-molybdenum weld metals intended for 2¼% chromium-1% molybdenum base metal are given on Fig. 3-6.

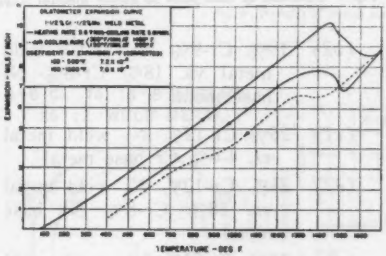


Figure 3

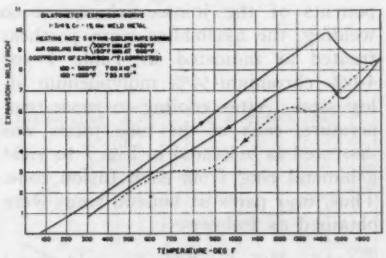


Figure 4

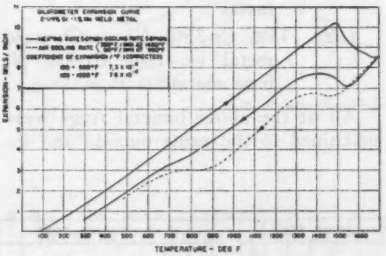


Figure 5

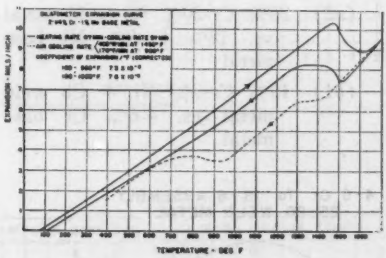


Figure 6

FERRITIC-AUSTENITIC WELDMENTS

The design of steam generation plants for operation at 1050° F. may present the problem of joining austenitic to ferritic tubing. The problem lies in the fact that there is a considerable difference between the coefficients of expansion of the austenitic and ferritic materials employed. Therefore, tests were performed to determine residual stress values in composite joints between austenitic and ferritic tubing. The object of these tests was to select the optimum electrode for the purposes intended on the basis of minimum residual stress conditions in the weldment. For these

tests, 4-6% chromium-½% molybdenum and 18% chromium-8% nickel tubing were employed to form four composite joints which were welded with each of the following electrodes:

(*a*) 25% chromium-20% nickel
(*b*) 19% chromium-9% nickel
(*c*) 25% chromium-12% nickel
(*d*) 15% chromium-35% nickel

Prior to welding, these assemblies were preheated 600° F. on the ferritic side and 400° F. on the austenitic side of the joint. These temperature levels represent the inverse ratio of the coefficients of expansion of the components of the joint. Subsequent to welding, the assemblies were postheat treated as indicated in Table 1 for 4-6% chromium-½% molybdenum alloy steel. After cooling to room temperature, each of the tube joints was dissected as indicated in Fig. 7 to yield a bimetal ring from each fusion zone. Thus, four pairs of bimetal rings were obtained as follows:

(*a*1) 25% Cr-20% Ni weld metal vs. 4-6% Cr base metal
(*a*2) 25% Cr-20% Ni weld metal vs. 18% Cr-8% Ni base metal
(*b*1) 19% Cr-9% Ni + Cb weld metal vs. 4-6% Cr base metal

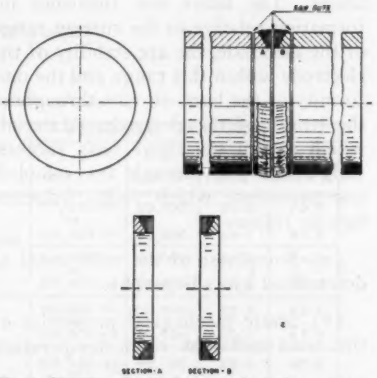


Fig. 7 Dissection of tube joint to obtain bimetal rings

(*b*2) 19% Cr-9% Ni + Cb weld metal vs. 18% Cr-8% Ni base metal
(*c*1) 25% Cr-12% Ni weld metal vs. 4-6% Cr base metal
(*c*2) 25% Cr-12% Ni weld metal vs. 18% Cr-8% Ni base metal
(*d*1) 15% Cr-35% Ni + Cb weld metal vs. 4-6% Cr base metal
(*d*2) 15% Cr-35% Ni + Cb weld metal vs. 18% Cr-8% Ni base metal

All of the above bimetal rings were machined to the same outside and in-

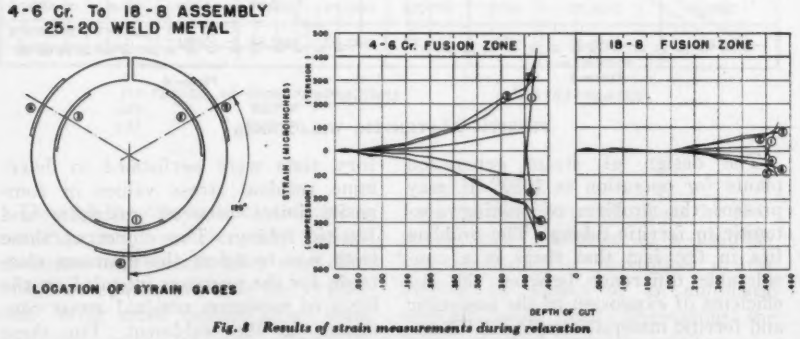


Fig. 8 Results of strain measurements during relaxation

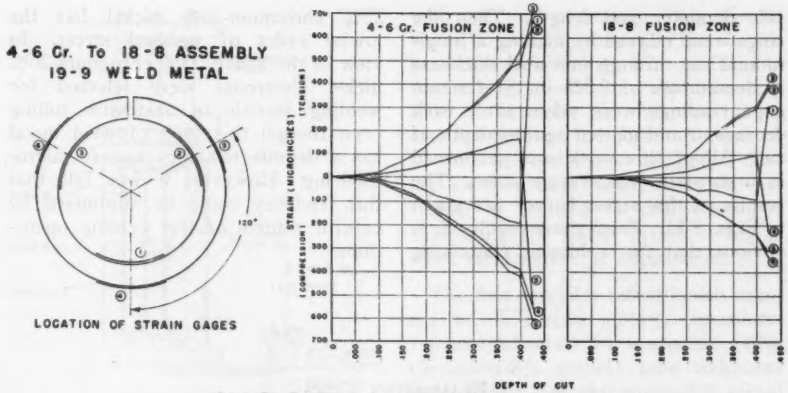


Fig. 9 Results of strain measurements during relaxation

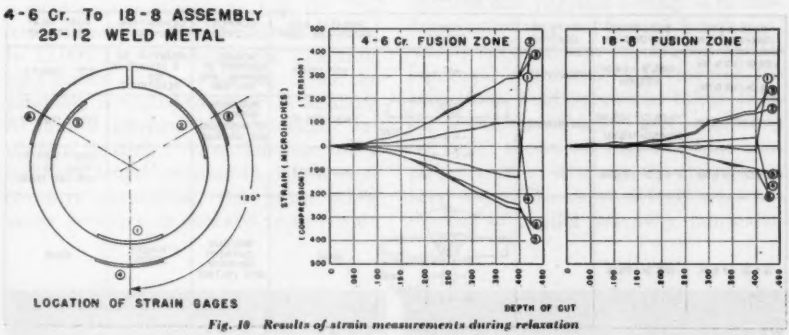


Fig. 10 Results of strain measurements during relaxation

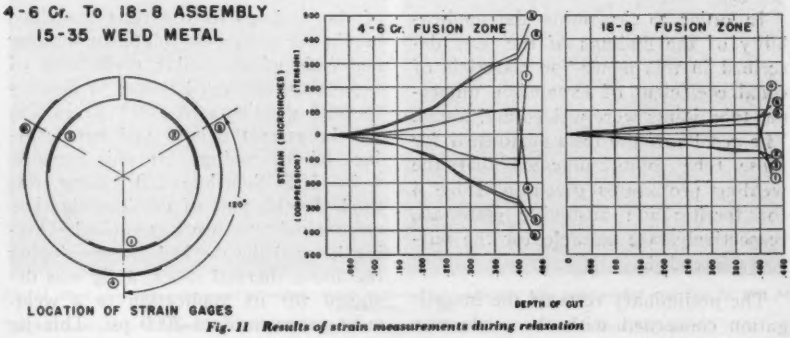
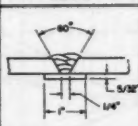
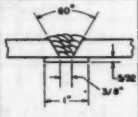


Fig. 11 Results of strain measurements during relaxation

side diameter and length. Then the rings were relaxed by making a longitudinal cut through one wall thickness in decrements of 0.025 in. Sr-4 strain gage readings were taken after each decrement and plotted against depth of cut. All of this work was performed in a constant temperature room. The results of this stress survey are given in Figs. 8-11. From these results, it is obvious that the weldment containing 25% chromium-20% nickel has the lowest order of residual stress. In view of the above, 25% chromium-20% nickel electrodes were selected for welding ferritic to austenitic tubing even though this type of weld metal has a definite tendency toward microfissuring. However, it was felt that this tendency could be minimized by careful control of the welding operations.

Evaluation Tests

Table 4—Welding Procedures

BASE METAL COMPOSITION	ELECTRODE COMPOSITION	TYPE OF COATING	JOINT DESIGN	PREHEAT AND INTERPASS TEMP.	CURRENT LEVEL	DEPOSITION TECHNIQUE	POST HEAT TREATMENT
1/2% Cr - 1/2% Mo	1/2% To 1-1/4 Cr - 1/2% Mo.	CELLULOSE		400-500° F	NORMAL CURRENT AT NORMAL ARC VOLTAGE	WEAVE UP TO 3 DIAMETERS OF ELECTRODE	1275 - 1300° F.
1% Cr - 1/2% Mo							
2-1/4% Cr - 1% Mo	1-1/2% Cr - 1/2% Mo To 2-1/4% Cr - 1% Mo.	LOW HYDROGEN		400 - 600° F.	MAXIMUM CURRENT AT MINIMUM ARC VOLTAGE	"	1300 - 1325° F.
4-6% Cr - 1/2% Mo	4-6% Cr - 1/2% Mo	"		500 - 600° F	"	"	1300 - 1325° F. COOL TO 900 AND HOLD
18% Cr - 8% Ni.	18% Cr - 8% Ni.	"		NONE	MINIMUM CURRENT MINIMUM ARC VOLTAGE	STRINGER BEADS	NONE

In order to determine the applicability of the findings of the tests described in this paper for materials of equal coefficient of expansion, numerous tube joints were welded and tested. The results of the tests performed for these tube joints indicated that the welding procedures given in Table 4 for ferritic and austenitic materials, respectively, are suitable for the purposes intended.

The preliminary tests of the investigation concerned with the evaluation of the findings of the tests described herein for composite joints, comprising materials of dissimilar coefficients of expansion, employed means to develop thermal gradients of 800° F. in 2½ sec. between the bore and outer surface of the tubing. For this purpose, 4 in. IPS Schedule 160 tubing was used. In this part of the investigation, no internal pressure was applied. Having learned the method for developing the above thermal shock, a jig was designed for its application to a weldment pressurized to 2000 psi. This jig

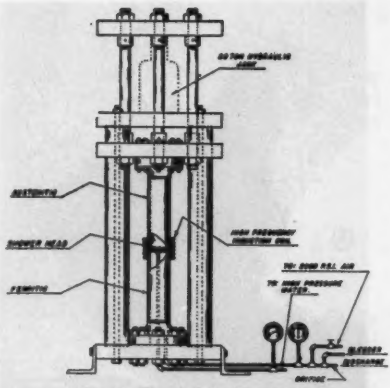


Fig. 12 Thermal shock apparatus

is shown schematically on Fig. 12. The test assembly is pre-stressed in tension to 12,000 psi. employing the hydraulic jack shown on top of the jig. Then the assembly is heated to 1050° F. in about 20 min. employing high-frequency induction heating. After holding for a sufficient length of time to obtain temperature saturation, the joint, while under pressure, is shocked from within with one pint of water. This pint of water, injected by means of the shower head shown on Fig. 12, produces a thermal gradient of approximately 800° F. between the outer periphery and the bore of the joint. A total of 20 such thermal shocks is applied during an 8-hr day. Thereafter, the assembly is allowed to cool overnight developing the initial preset stress as room temperature is approached.

The first assembly tested in this manner was a ferritic (2¼% chromium-1% molybdenum) vs. austenitic (18% chromium-8% nickel) tube weldment containing a 25% chromium-20% nickel weld. After 300 thermal shocks, the assembly was removed for surface inspection and physical testing. The surface inspections performed for the assembly revealed nine fine cracks. Metallographic examinations performed for transverse weld specimens taken from the assembly revealed root cracks of the order shown on Fig. 13. This condition was evident in the results of bend tests. The above defects were not evident in a pilot assembly fabricated

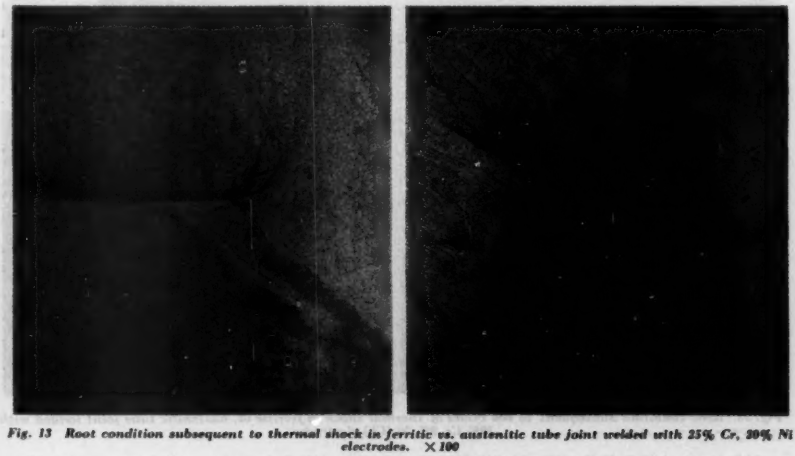


Fig. 13 Root condition subsequent to thermal shock in ferritic vs. austenitic tube joint welded with 25% Cr, 20% Ni electrodes. ×100

(a) Backing ring 2¼% Cr, 1% Mo; (b) ferritic 2¼% Cr, 1% Mo; (c) weld metal 25% Cr, 20% Ni and (d) austenitic 18% Cr, 8% Ni

employing identical welding procedures. The latter assembly was tested without being exposed to thermal shocks.

In view of the above mentioned defects, a second 2¼% chromium-1% molybdenum vs. 18% chromium-8% nickel tube weldment was subjected to the thermal shock test described above. This weldment was fabricated employing 19% chromium-9% nickel plus columbium electrodes. After 300 thermal shocks, this assembly was found to be free of surface defects. However, a small transverse weld specimen removed by a weld prober was found to contain the root condition shown on Fig. 14. This condition has the appearance of a shrinkage cavity and is apparently local in character as the other side of the specimen was free of this condition. The distance between the two faces of the specimen examined was approximately ¼ in.

The volume removed by the weld prober from the above joint was replaced by welding and the assembly re-installed in the thermal shock jig.

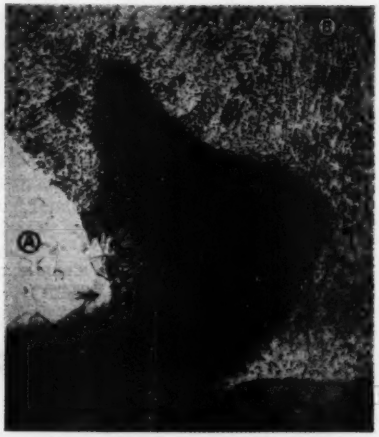


Fig. 14 Root condition subsequent to 30 cycles of thermal shock in ferritic vs. austenitic tube joint welded with 19% Cr, 9% Ni (Cb) electrodes. × 100

(a) Austenitic base 18% Cr-8% Ni (b) weld metal 19% Cr-9% Ni (Cb)

This assembly was subjected to another 300 cycles of thermal shock. At this time the root condition shown on Figure 15 was found. This condition was substantiated by the results of the root bend test.

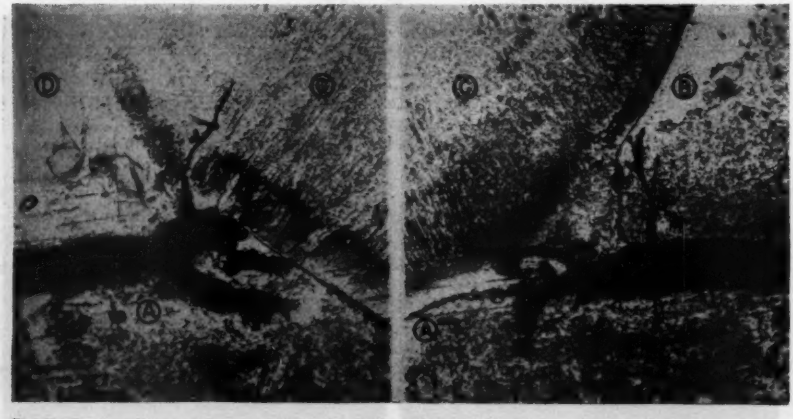


Fig. 15 Root condition subsequent to 600 cycles of thermal shock in ferritic vs. austenitic tube joint welded with 19% Cr, 9% Ni (Cb) electrodes

(a) Backing ring 2¼% Cr, 1% Mo; (b) ferritic 2¼% Cr, 1% Mo; (c) weld metal 19% Cr, 9% Ni (Cb) and (d) austenitic 18% Cr, 8% Ni

Summary

The above described program resulted in the development of welding procedures for the following alloy steel tubing wherein each alloy is welded to itself:

(*a*) ½% Cr-½% Mo
(*b*) 1% Cr-½% Mo
(*c*) 2¼% Cr-1% Mo
(*d*) 4-6% Cr-½% Mo
(*e*) 18% Cr-8% Ni plus Cb

Tests of composite joints between 2¼% chromium-1% molybdenum and 18% chromium-8% nickel steels wherein thermal shock is employed indicate that 19% chromium-9% nickel (Cb) weld metal is apparently more suitable for the purposes intended than 25% chromium-20% nickel. However, final conclusions concerning the suitability of these composite joints for use in high-pressure, high-temperature steam systems wherein the frequency and magnitude of thermal shock conditions are of prime importance, are being held in abeyance pending completion of considerably more research work in connection with this problem.

The author is indebted to Bela Ronay, Superintendent of the Welding and Electrical Laboratory at the U. S. Naval Engineering Experiment Station for his constructive review of this paper.

FUNNEL DESIGN AND SMOKE ABATEMENT

ACKNOWLEDGMENT

This article, a joint paper read before The Institution of Naval Architects and The Institute of Marine Engineers in London, on April 25, 1950, was published in the November 1950 issue of the "Institute of Marine Engineers." It is divided into two parts. The first part, by E. OWER of the British Shipbuilding Research Association, deals with the causes of smoke pollution and discusses several possible methods of prevention. The second part, dealing with the experimental method by which the problem was studied, was written by C. H. BURGE, of The National Physical Laboratory.

Part I

THE CAUSES OF SMOKE POLLUTION AND POSSIBLE METHODS OF PREVENTION

INTRODUCTION

There is little doubt that aesthetically the modern ship, with its short, raked, streamlined funnels, is much more satisfying in appearance than the ships of some twenty or thirty years ago, in which high, spindly, cylindrical funnels were the general rule. There is equally little doubt that this change in funnel design is mainly responsible for the trouble that occurs so much more frequently today, namely the descent of fumes, smoke and soot on to the deck.

The problem has received a good deal of attention during the last ten years or so, both for naval and merchant ships; and, in July of last year, the British Shipbuilding Research Association issued a survey of existing knowledge on the subject to its member firms. This survey was prepared by the author of Part I of this paper, who has been given permission by the Association to make use of it here.

The cause of the trouble is essentially aerodynamic in nature and associated with the flow of air over the superstructure of the ship and around the funnel. Many cures have been suggested, some partly successful, others not; and it is true to say that no successful one has yet been devised which is free from all objections. It is, indeed, difficult to see how such a remedy can be found, short of a reversion to the high, cylindrical funnels of the past.

GENERAL STATEMENT OF THE PROBLEM

In general, the funnel exhaust is partly gaseous and partly solid. The solid matter, consisting of particles of soot or ash large or dense enough to have an appreciable velocity of descent in air, provides a special problem about which very little can be done, except to prevent it from being formed or to remove it from the exhaust before it is discharged into the air. Once these particles are ejected, they must fall on to the deck, if their natural velocity of fall is high enough to cause them to drop the necessary distance before they are carried clear of the ship by whatever horizontal speed they can pick up from the relative wind. The special case of these larger particles is discussed later; the main problem is that of the gaseous products of combustion, and of the small particles forming the smoke proper whose behavior can be considered for the present purpose as essentially the same as that of the gases.

The path of the smoke trail from a funnel *of given shape and size* is determined mainly by four variables, namely:

(*a*) The speed of the relative wind.

(*b*) The speed of emission of the smoke.

(*c*) The density (weight per unit volume) of the smoke.

(*d*) The form of the superstructure ahead of the funnel, and, to a smaller degree, the form of the bows and the superstructure aft.

Strictly, another variable should be added to this list, that is the direction of the relative wind, but this can be regarded as equivalent to a change in variable (*d*), and so can be discussed under that head.

It should be repeated that these four variables relate only to a funnel of given shape and size; any change in the form of the funnel introduces another important variable which will be considered later. Associated with it is again the effect of a change in the direction of the relative wind, which, except for a vertical, cylindrical funnel with an unraked top (for which there is no such effect), can be regarded as equivalent to a change in shape of the funnel.

Restricting ourselves for the time being to a particular funnel, that is, neglecting the effect of changes in funnel shape, we can perhaps most easily form a conception of the behavior of the smoke trail by considering firstly the simplest case in which only variable (*a*) operates, and then showing how the other variables modify this behavior. We start, therefore, with a funnel rising from a flat deck, and assume no obstructions for a considerable distance forward (i.e., negligible effect of bows and superstructure), small velocity of emission of the smoke, and density of smoke not appreciably different from that of the atmosphere. Even in a slight wind, the smoke trail would then be horizontal,* and, were it not for the low-pressure region behind the funnel, the lower edge of the trail would be level with the top of the funnel. But this low-pressure region causes the smoke to descend below the funnel top: in effect, as far as the smoke trail is concerned, the funnel top is somewhat below the actual top. This is shown by the lighter smoke trails in Fig. 1, which is a photograph of the smoke from a group of tall, cylindrical power-station chimneys. There is evidence also of the effect of eddies shed by the chimneys, which cause occasional puffs of smoke to descend well below the main trail. Further reference to Fig. 1 will be made later under the heading of funnel shape. For the present, we need only

* The relatively small spread due to natural diffusion is neglected.

remember that the funnel behaves as though its true or "effective" height were less than its real height, and we can neglect the effect of the low-pressure region behind the funnel if we speak of effective top or effective height instead of real top or real height.

We see then that, for the simple conditions postulated (i.e., no obstructions ahead and negligible velocity of emission and differences of density), the lower edge of the horizontal trail from the effective funnel top represents a boundary below which the smoke does not fall appreciably; but any departure from these conditions will alter the path of the smoke trail. Thus, the path will obviously be raised by the upward momentum of the funnel gases if there is any appreciable velocity of emission [variable (*b*)]; and, because of the relatively high temperature of the gases [variable (*c*)], their density is in practice always below that of the air, and this again raises the trail.

That the effect of temperature can be quite substantial is shown by the difference between the dark and light smoke trails in Fig. 1. The light, dense trails come from chimneys emitting "washed" smoke from which most of the solid matter has been removed, and the efflux consists very largely of condensed steam, which is considerably cooler than the dark, unwashed smoke coming from the third chimney. It will be seen that the darker smoke trail is distinctly higher, despite the fairly high natural wind that was blowing at the time the photographs were taken.

FIG. 1

Thus we see that, of the four variables, (*a*) by itself produces an approximately horizontal trail, while (*b*) and (*c*) will always act in the sense of an improvement by causing the gases to rise, if even slightly, above this horizontal limit. Whenever trouble occurs from the descent of smoke and gases abaft the funnel, it is due† to variable (*d*), i.e., the form of the bows and superstructure, particularly in its relation to the position of the effective funnel top. How this occurs is explained in the next section.

THE EFFECT OF OBSTRUCTIONS

When a wind strikes a bluff obstacle projecting from a flat surface, it is deflected upwards by the top edge in the manner sketched in Fig. 2. Above a certain line A B C, the air continues to flow smoothly, almost as though A B C were the boundary of the bluff

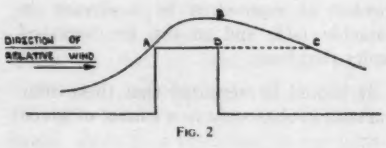


FIG. 2

† Apart from the effect of funnel shape, for which see later.

obstacle. Eventually, the line A B C returns to the level of A D, the top of the obstacle, and then falls below it if A D is shorter than A C; but this occurs at a distance from A which, in cases of practical interest, is several times the height of the obstacle. The space A B C D between the line and the top of the obstacle is filled with violent eddies; and the line A B C therefore represents a boundary, which we shall call the turbulence boundary, between regions of smooth and disturbed flow. The course of the line A B C can be changed considerably by alterations in shape of the obstacle, particularly by relief of the sharpness of the top edge by means of curves or chamfering;[1] but, in general, whenever a bluff obstacle is exposed to a wind there is a region of eddying flow above it,, separated from the smooth flow by a fairly well-defined boundary.

Any structure, such as a deckhouse or bridge, erected on the deck of a ship meeting a head wind, constitutes a bluff obstacle from which a turbulence boundary springs. The bows themselves form another obstacle: if the wind is dead ahead and the bows are flared, the disturbance introduced into the flow is more lateral than vertical, and comparatively little eddying results over the deck forward of the superstructure; if, however, the wind is inclined, the bows act as a bluff obstacle and there may be large eddies over the deck.

If smoke from the funnel enters the general flow well above the turbulence boundary it will not descend to deck level before the boundary reaches that level; but if it is introduced below the boundary it will come within the influence of the eddies and may reach the deck anywhere aft of the funnel, or, indeed, even forward of it. The three photographs of Fig. 3, which have been provided by Mr. Burge, who describes the experimental technique in Part II, show the turbulent zone on a model ship. Fig. 3(*a*) shows the appearance of the smoke trail in a free stream; for this experiment the smoke was introduced well to the side of the model, and was clear of the disturbance it caused. For Fig. 3(*b*), the smoke was introduced in the central vertical plane just below the level of the turbulence boundary springing from the superstructure;

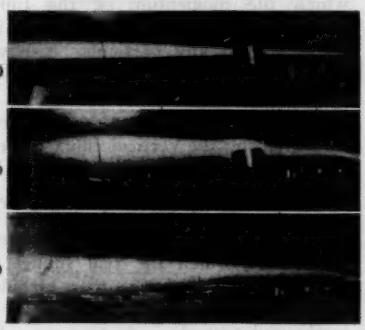


FIG. 3

the smoke is drawn down by the eddies and fills the space behind the funnel almost completely. Fig. 3(*c*) shows similar conditions with the funnel removed. It should be noted that the separate eddies apparent in the instantaneous-exposure photographs of the power-station chimneys (Fig. 1) are not seen in Fig. 3 because the latter photographs were given time exposures, as described by Mr. Burge.

It follows from the foregoing that the smoke must be discharged well into the smooth flow above the turbulent zone if we wish to be certain that the smoke will not descend to deck level before the turbulence boundary, which generally occurs well beyond the stern of the ship. To reach this favorable point of discharge we can either make the funnel sufficiently tall, which, in modern ships, may entail a considerable increase in height, or we can give the smoke from a low funnel enough upward momentum to carry it above the turbulent zone.

THE EFFLUX VELOCITY OF THE FUNNEL GASES

The upward momentum of the funnel gases can be increased for a given uptake diameter by increasing either their temperature or their velocity, but the first alternative is so much more wasteful in power than the second that it need not be considered. The amount by which the momentum of the funnel gases must be increased to effect an improvement in any particular case will obviously depend on the horizontal speed (v) of the relative wind. If, therefore, we increase the momentum by increasing the vertical efflux velocity (s) of the smoke, the significant quantity is the ratio s/v.

Numerous experiments have been made in this country mainly at the National Physical Laboratory, and by Nolan[2] in America, on the effect of varying this ratio. These experiments show that when the ratio is equal to about 1, the axis of the smoke plume from a cylindrical chimney discharging into free air is roughly horizontal, and that the plume is raised appreciably when the value of the ratio is about 2.

SHAPE OF FUNNEL

(*a*) *Wind Ahead.* The minimum amount by which the real funnel top must protrude above the turbulent boundary to ensure freedom from the smoke nuisance depends to a considerable extent on the shape of the funnel. It has already been stated that there is a low-pressure region behind the funnel, which causes the smoke to be drawn down below the top (see Fig. 1). The height depends largely on the shape of the funnel, as can be seen from the photographs of Fig. 4. The object in the upper photograph is a model cylindrical funnel emitting smoke in a wind tunnel, and it will be seen that the smoke is sucked down immediately behind the funnel to a distance of more than half the height. The difference between this photograph and those of Fig. 1 is partly due to the fact that the long duration of the exposure for the model funnel gives an exaggerated impression of the amount of smoke behind it at any given instant. The essential feature about this photograph, however, is the difference between it and the lower one, which was produced by placing a streamlined sheet-metal casing around the cylinder; the breadth of the cross-section of the fairing was just enough to enclose the funnel, and its length along the wind direction was about four times the breadth.

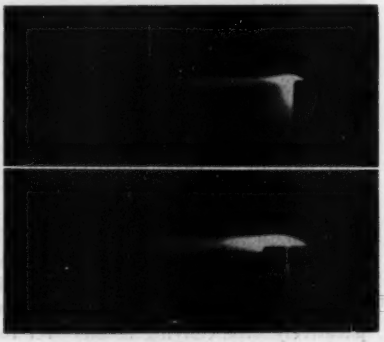


FIG. 4.—THE EFFECT OF A STREAMLINED FAIRING PLACED AROUND THE FUNNEL
Velocity ratio $s/v = 1/1$.

The great improvement due to the streamlining with wind ahead is apparent from these photographs: the intensity of the suction behind the funnel is much reduced and the smoke trail is correspondingly less depressed.

Another factor that influences to some extent the path of the smoke trail is the shape of the top of the funnel. We know, for example, that a raked top is bad. This is shown diagrammatically in Fig. 5, which gives the approxi-

mate paths of the smoke plume from a typical streamlined funnel under various conditions. In Fig. 5A the wind is dead ahead, and the effect of the rake is to cause an eddy as shown above the funnel and a break-away of the flow from the forward edge. The lines (*a*), (*b*), (*c*) and (*d*) represent approximately the paths that would be followed by streamers attached at the points from which the lines spring. In passing, we may note that experiments have shown that spaces between deckhouses lower the flow boundary, as indicated by line (*c*).

(*b*) *Yawed Winds.* With the conventional funnel with raked top, the effect of an angle of yaw limited to 10 deg. is shown in Fig. 5B, as compared with Fig. 5A. The main volume of the smoke is confined within the region marked "Apparent Main Plume," but a perceptible amount of the efflux breaks away and diffuses over the forward edge of the funnel—quite irrespective of the position of the uptake within the funnel casing—and ultimately follows the course between the main plume and the dotted boundary.

If the rake is removed, so that the funnel has a flat top, conditions are greatly improved, as shown in Fig. 5C. The flow lines are here drawn for the design of funnel in which the uptakes do not occupy the whole of the funnel space. Again the diffused smoke boundaries are shown dotted.

The addition of a flat plate in the form of a collar around the funnel top ensures that the point of break-away remains at the top of the funnel, and very little smoke diffuses over the edge of the funnel, as shown in Fig. 5D, provided that the yaw does not exceed about 10 deg.

When the angle of yaw is greater than about 10 deg., its effect can become very large. Experiments show that the worst conditions occur when the

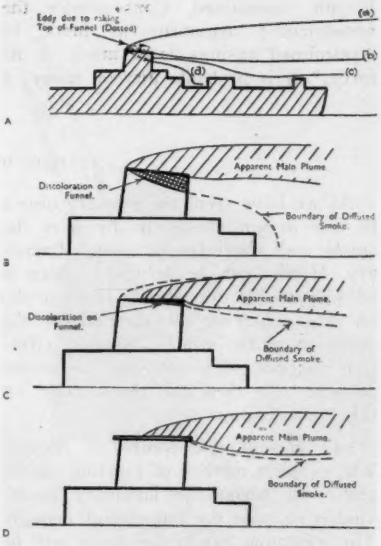


FIG. 5

angle of yaw is between 20 deg. and 30 deg. At these angles, the suction in the lee of the average streamlined casing is very high—probably at least as great as that behind a cylindrical casing—and it acts over an extended area. In these conditions, although the main plume appears to take a path clear of the ship, a considerable volume of loose smoke spills over below the main plume and descends vertically into the low-pressure region in the lee of the funnel, reducing the effective height of the latter very considerably. This loose smoke afterwards disperses down the lee side of the ship and pollutes those parts of the deck and superstructure that lie in its path.

It follows from the foregoing that in winds inclined by more than about 10 deg. a streamlined funnel is in fact no longer streamlined. Hence ships will very often be operating in wind conditions in which the great improvement achieved by streamlining in head winds

is not maintained. Consequently the aerodynamic argument in favor of streamlined casings loses much of its force:* it is probable that on many, if not most, occasions a cylindrical casing of the least diameter necessary to deal adequately with the flue gases would be better.

METHODS OF PREVENTION

As we have seen, the primary object in any design should be to eject the smoke well above the turbulence boundary. If this can be achieved, there is unlikely to be any trouble from smoke on deck whatever the strength or the direction of the wind. The most effective methods of preventing the smoke nuisance are therefore those based on this consideration.

(*a*) *Height and Shape of Funnel.* The simplest method of ejecting smoke above the turbulence boundary is obviously to make the funnel high enough. The minimum height necessary will be the height of the turbulence boundary, plus the distance by which the suction due to the flow of air past the funnel draws the smoke down below the top. In all except head winds and winds within a few degrees of the ahead (or astern) direction, this distance will be least for a cylindrical funnel of the minimum diameter necessary to deal adequately with the boiler gases. We are therefore led to the conclusion that a return to the tall, narrow, cylindrical funnel offers the cheapest and the most effective cure. If, nevertheless, streamlined funnels are retained for reasons of appearance, the length/breadth ratio of their cross-sections should be as low as possible, their tops should be horizontal rather than raked, and if an annular collar around the top can be tolerated some improvement is to be expected from it.

Improvement may also often be obtained by lengthening a funnel by less than the full amount shown by tests to be desirable for the worst conditions. Thus in a number of warships tested for the Admiralty, increases in funnel height by about 1 to 1½ mean diameters were found to be sufficient. It is not possible, however, to give any general indication of the minimum increase in height that will suffice: if the designer wishes to be certain that any particular ship will be free from smoke trouble, his safest course will always be to have a model tested.

(*b*) *Efflux Velocity of the Funnel Gases.* Apart from increasing the funnel height, the method that offers the most promise—although at the expense of some extra power—is to increase the efflux velocity of the funnel gases, either by reducing the exit area of the uptake or by blowing additional air out of the funnel. As already stated, the effect of increasing the efflux velocity usually becomes noticeable when the value of the ratio s/v is between 1 and 2. This enables us to form some estimate of the order of funnel uptake velocity that will be required in practice. For example, if we consider a ship steaming at 20 knots into a head wind of 20 knots, the efflux velocity required for $s/v = 1$ is about 4000 ft. per minute, and for $s/v = 2$ about 8000 ft. per minute. If the volume of the funnel gases is 30,000 cu. ft. per minute, and their temperature about 350° F., the values of the horsepower lost in the efflux corresponding to these velocities are 3 and 12, and the values of the extra fan head required are 0.6 and 2.6 of water.

These figures for the power lost in the exhaust and the increased duty re-

* It is assumed that a swivelling streamlined casing is impracticable.

FIG. 6

quired of the fans are quite small percentages of the total demands of the ship, which have in any case to be provided; and in view of the valuable improvements the effect in the smoke problem it appears well worth providing for them in the design stage.

An example recently occurred in which effective use was made of both increased height and increased gas velocity, with a very considerable improvement in conditions at small cost. Fig. 6 shows the smoke trail observed on the model ship in the wind tunnel at wind ahead. It will be seen that the trail sweeps a considerable portion of the deck, which was confirmed by full-scale observation on the ship itself. As a result of the wind-tunnel experiments, the diameter of the uptake was halved by fitting a reducing cone with a total angle of rather less than 20 deg. The cone projected about 9 ft. above the funnel top, and the velocity of the funnel gases at efflux was about 6000 ft. per minute at full power (8000 H.P.). The corresponding speed of the ship was 17½ knots, so that in a 20-knot head wind the value of s/v was 1.6. With this modification, it was reported that no smoke trouble was experienced. Figs. 7 and 8 show photographs taken on the ship at sea; for Fig. 7 smoke was specially made, and when Fig. 8 was taken tubes were being blown. The improvement compared with Fig. 6 is very marked; and it was achieved at a cost of just over 6 H.P. extra in the discharged gases and an extra 1½ in. of fan head.

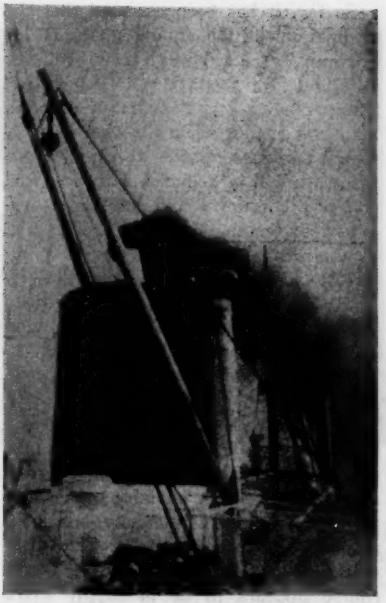


FIG. 7

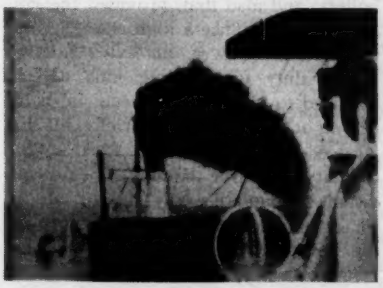


FIG. 8

The use of a projecting cone, as in this case, does not only increase the funnel height and the efflux velocity; it has the additional advantage that, since it is considerably smaller than the funnel casing, the suction behind it is less and more localized. Hence the effective height of the funnel is increased by more than the added height of the cone.

So far, it has been assumed that the momentum in the funnel gases is increased by increasing their velocity only, not by providing additional air and so increasing their mass as well. This means that when the ship is operating at reduced power the velocity of the gases will be reduced, since less fuel is being consumed and less air is required for combustion. In the ship just mentioned, the values of s/v in a 20-knot wind at 16½ knots (full speed), 15 knots and 13½ knots are respectively about 1.6, 1.4 and 1.0. The relative wind over this range only falls from 36½ to 33½ knots. From this it is clear that if only the combustion air is used, although it may be speeded up sufficiently at full power by reducing the uptake diameter, conditions will not be so good at lower powers. A variable outlet area would overcome this difficulty, but probably only at the expense of other difficulties of a different kind. A better course appears to be to supply extra air. It is important to notice that this extra air need not pass through the boilers, and also that, provided the fans are able to produce a high enough value of s at full power, no extra fans or higher duty for existing fans will be required to maintain the requisite value of s/v at lower powers. Briefly, all that is needed is to run the fans at full capacity whenever the natural wind conditions demand it; only the air needed for combustion need be passed through the boiler, the air trunking being arranged to bypass the remaining air direct to the uptake. This possibility was first mentioned by Mr. Burge, who discusses it in greater detail in Part II of this paper.

An alternative way of using extra air is to inject it in the form of a high-velocity sheath or annulus surrounding the natural funnel gases. Nolan discussed this possibility in his paper and said that it was "very effective. . . .

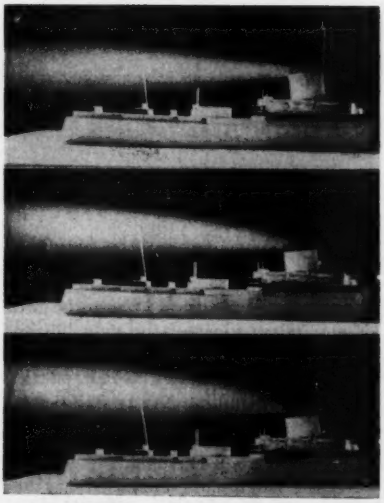


FIG. 9.—THE EFFECT OF AN ANNULAR AIR JET AROUND A FUNNEL UPTAKE IN THE CENTRE OF THE FUNNEL CASING
Velocity ratio s/v = 1/2.
Top—Normal funnel.
Centre—Normal height reduced 9 ft. Jet velocity = 2·5 v.
Bottom—Normal height reduced 9 ft. Jet velocity = 4·0 v.

The annulus acts as an ejector and also an isolating belt which helps to prevent the stack gases from being drawn into the (low-pressure) zone immediately behind the stack." The effectiveness of this high-velocity sheath is shown by the photographs of Fig. 9, which were taken at the National Physical Laboratory. A marked improvement in the path of the smoke trail is produced even with a funnel of *reduced* height and with the low s/v ratio of ½.

So far all the experimental data on this device have been obtained on the model scale, and their practical implications are discussed by Mr. Burge in Part II. The correlation between model and full scale has yet to be established.

(*c*) *New Shape of Funnel.* In May 1948, an interesting new development was described in a paper read by Valensi and Guillonde[3] to the Associa-

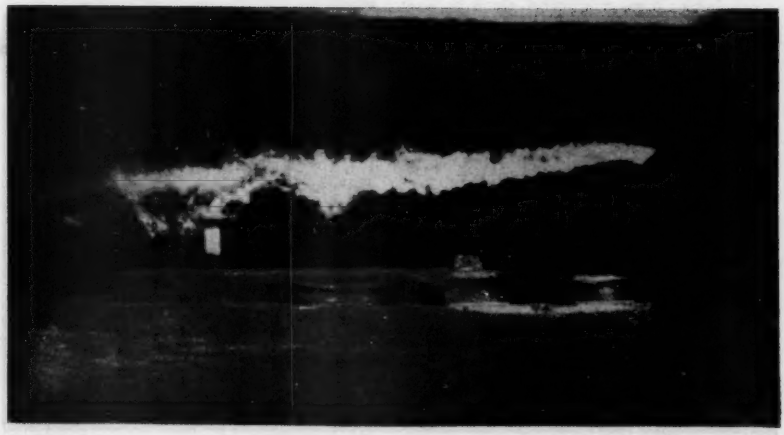


FIG. 10

tion Technique Maritime et Aéronautique. This takes the form of a funnel whose cross-section is shaped like that of a symmetrical aerofoil with maximum thickness 20 per cent of the chord. In side elevation, the funnel resembles a short half-wing of an aeroplane, with a rounded tip forming the funnel top (see Fig. 10). The smoke emerges in the locality from which the trailing wing-tip vortex springs, and is carried downstream by this vortex, one of the chief characteristics of which is that it preserves its individuality for a considerable distance downstream. The authors of the paper state that they developed this device in wind-tunnel experiments and confirmed its effectiveness in full-scale tests at sea. In the particular case that they examined they found that, provided the funnel was of sufficient height just to penetrate the turbulence boundary, the smoke was carried well clear of the deck at all inclinations of the relative wind from dead ahead to 30 deg. on the bow. A typical condition is shown in Fig. 10, which shows their model in the wind tunnel in a 10 deg. wind. Fig. 11 shows the same model with a conventional funnel; the great improvement is evident. The paper itself includes other photographs taken in the wind tunnel at zero yaw. Again the improvement is pronounced, and it is confirmed by a photograph of the full-scale vessel taken at sea.

From the information given in the paper, this device, which is the subject of French Patent No. 537822 of 12th July, 1947, seems to be one of considerable promise. Further verification of its effectiveness appears desirable, particularly since, so far, it appears from the paper to have been tried only in one particular ship. Its behavior in following winds also needs examination.

(*d*) *Deflector Plates.* Experiments made for the Admiralty have shown that inverted aerofoils placed ahead of the funnel top and at suitable angles of incidence (see Fig. 13) can produce a powerful up-draught at the top of the funnel, which will carry the smoke well above the turbulence boundary. Similar

FIG. 11

FIG. 12

FIG. 13

effects can be obtained with flat plates; but in either case the deflectors will probably have to be at a height at least equal to that of the turbulence boundary, and are unlikely to be effective if the funnel top is much below it. Further, the deflectors must be fairly large —of span about equal to the funnel width and chord about one-third or one-quarter of this width—and strongly mounted, since the wind forces on them may be substantial. This need for size and strength may be a disadvantage.

There is, however, no doubt that the device is effective. Fig. 12 is a photograph of the smoke from the funnel of a destroyer, showing the after armament enveloped in smoke. Figs. 13 and 14 show how the smoke is lifted by a pair of aerofoils at 45 deg. incidence, and how the trail keeps well clear of the deck. All the photographs relate to a ship speed of 20 knots and a relative wind of 42 knots. Fig. 15 is a sketch of the dimensions and the relative positions of the funnel top of the same vessel, and of a pair of flat-plate deflectors designed on the basis of the results with aerofoils depicted in Figs. 13 and 14.

The deflector plates must be upstream of the funnel; hence an additional pair will be necessary for following winds,

FIG. 14

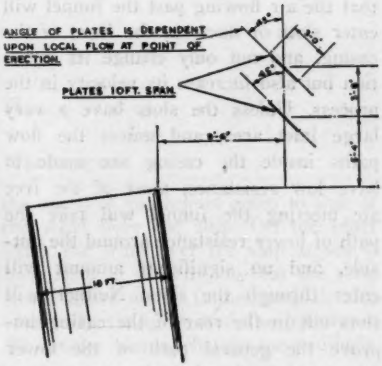


FIG. 15

unless the pair used for head winds is mounted on a movable support. Adjustments of the deflectors, or additional deflectors, will also be required for winds yawed by more than 5 deg. or so.

(*e*) *Cowls and Hoods.* A cowl or a hood on the top of a funnel sometimes effects an improvement. One of the main uses of such devices is to improve local conditions; a typical example is that of a bridge situated fairly near the funnel top, where trouble may be experienced from smoke in a following wind. Two classes of naval vessels in which this occurred were reported to be improved by funnel cowls, but in each case the funnel height was also increased. In any event, fitting a cowl to the top of an existing funnel in itself obviously increases the effective height, so that some of the improvement must be due to this cause as well. Some may also be due to the fact that a cowl generally rounds off the top of the casing and eliminates the sharp forward edge.

Devices of this kind do not lift the smoke as do some of the more effective expedients already mentioned; they merely direct it away from the region which is to be kept clear. It is probable that their usefulness is restricted to special cases such as that mentioned above. In winds off the bow they are better than conventional funnels in that the smoke is emitted in a region further away from the leeward suction and so is less likely to be drawn down. On the other hand, backward-facing cowls remove all or most of the vertical velocity from the smoke before emission, and are subject to obvious disadvantages in following winds.

To reduce extra loads on the fans, cowls should be so designed that any change in the direction of flow of the gases from vertical to horizontal occurs as gradually as possible.

(*f*) *Miscellaneous Devices.* During the past few years a number of aerodynamic devices have been introduced by different designers. In general, we can say of such devices that, if they are to be effective, they must either divert the main flow, or else expel the smoke from the funnel at a velocity high enough to ensure penetration to a height above that part of the flow which descends to deck level. In either case they must use the energy of the wind, and therefore they are unlikely to be effective if they offer appreciable resistance to the free flow. For this reason, little success has attended devices based upon the hope

that the air flowing past the funnel will enter slots or ducts in the front of the casing, and not only change its direction but also increase its velocity in the process. Unless the slots have a very large inlet area, and unless the flow paths inside the casing are made to have low resistance, most of the free air meeting the funnel will take the path of lower resistance around the outside, and no significant amount will enter through the slots. Neither will slots cut in the rear of the casing improve the general path of the lower fringes of the flow boundary because, as shown in Fig. 5B, the break-away at the top of the funnel occurs well forward.

Some of the modifications that have been introduced from time to time raise the point of emission above that from the original funnel. They are therefore equivalent to an increase in the height of the funnel; and if they produce any improvement at all, it is probably due mostly, if not entirely, to this increase in height and not to the modifications themselves. Moreover, even if there is some improvements at wind ahead, it does not follow that there will be any in a yawed wind, in which conditions are more stringent.

SOOT AND ASH

As already remarked, if comparatively large particles of soot and ash are emitted from the funnel, they will inevitably fall on to the deck if the speed of the relative wind is not great enough to carry them clear of the ship before their natural velocity of fall brings them down to deck level. This trouble is therefore most prevalent in harbor, in light winds and at slow operating speeds, or in following winds. The only possible remedies are to prevent the formation of these large particles as far as possible by constant attention to the combustion control, or to remove them before the smoke leaves the uptake by means of devices such as vortex soot extractors. It is a matter for the shipowner to decide between the nuisance of soot and ash contamination on the one hand and the initial cost and maintenance expense of soot extractors and combustion control on the other. It is, however, worth bearing in mind that speeding up the funnel gases, as advocated earlier in this paper for keeping the decks clear of gases and fine smoke, will help towards reducing trouble due to the coarser particles, because the particles will have further to fall and hence a better chance of being swept clear of the ship by whatever wind there is.

The coarse particles are also liable to be troublesome when boiler tubes are blown, but judicious adjustment of speed and course during the process, in relation to the strength and direction of the wind, can improve conditions considerably. The object should be either to get a high value of s/v, a sufficiently high value of v to carry the particles away, and a wind inclination of not more than 10 deg. ahead or astern; or a high value of s/v, a sufficiently high value of v, and a wind inclination of between 45 deg. and 90 deg., so that as much as possible of the soot is taken over the side. If the first alternative is adopted, the ship's speed will generally have to be reduced in a head wind to increase s/v, and may sometimes with advantage be increased in a following wind to get a sufficiently high v.

Part II

Experimental Technique

The direct purpose of wind-tunnel experiments on model ships is to facilitate the study of the flow pattern of funnel gases, firstly in relation to the estimated conditions of normal operation pertinent to the ship, and secondly to devise means of diverting the flow boundaries which descend to deck level within the dimensions of the ship. Various methods have been employed in attempting to raise the lower boundaries of the smoke plume clear of the decks and superstructure. Most of these have proved abortive, because they are only effective over a limited range of wind variation and, without the aid of power, they are unable to assist the dispersal of funnel gases. There is no empirical law which will enable the shipbuilder to arrive at the efficacy of a funnel auxiliary and he must of necessity have recourse to the method of trial and error. It follows therefore that, provided the results of model experiments may be relied upon, the wind tunnel is a more economical method of studying smoke conditions than direct full-scale investigation.

In the field of marine engineering, the wind tunnel is an unusual piece of research equipment, and the shipbuilder might, at first glance, have some misgivings in accepting the photographic records of the flow patterns given by the model, since they have the general appearance of being much better defined than those experienced full-scale. This apparent difference between the two sets of conditions is explained by the fact that the wind speed over the model is maintained at a constant value and the resultant smoke plume maintains a definite pattern according to the ratio of the speed of the funnel gases to that of the wind over the ship. On the other hand, the wind over the ship at sea is inconstant in velocity, so that the ratio to the speed of the funnel gases is also constantly changing and the pattern of the smoke plume changes accordingly.

The full-scale plume recorded by the eye is similar, in effect, to a series of instantaneous exposures; but in model experiment it would not be practical to define the plume boundaries by a sequence of instantaneous exposures, and therefore a time exposure is employed, usually of about 30 sec. duration. This record may be considered an integration of the sequence of instantaneous exposures over that period, and those wisps of smoke which occasionally break away from the apparent main plume actually define the boundaries of the diffused smoke. This diffused smoke is not as heavily laden with soot particles as the main body of the effluent, but over a period of time it will deposit a coating of soot on all structures within its path. The experimenter therefore assesses his success by the extent to which he has diverted this smoke from the decks and supertructure of the model.

In the course of the model experiments the comparison of several modifications may be required, and it is convenient to use the same velocity ratio at all angles of wind between 0 deg. and 30 deg., this being the usual range of the investigation. In the assessment of the final results, however, allowance must be made for the reduction in the speed of the relative wind as the ship turns across the natural wind.

EXPERIMENTAL APPARATUS

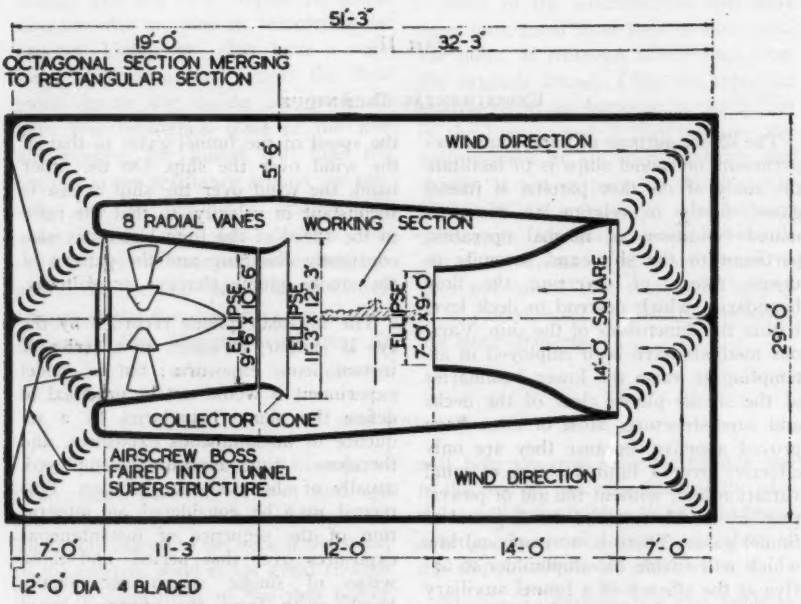


FIG. 16.—VERTICAL SECTION OF N.P.L. OPEN JET TYPE WIND TUNNEL

The Wind Tunnel. A longitudinal section of the wind tunnel used at the N.P.L. for experiments on model ships is given in Fig. 16.

The wind tunnel is an apparatus for generating a steady, uniform current of air, in which details of the flow past stationary models may be investigated, or, when necessary, the forces and pressures due to the flow can be measured. Fundamentally, the principles of wind-tunnel research as applied to investigations of the air flow over ships are simple, the flow being examined generally by means of smoke introduced at suitable places. Considerable care, however, must be taken if the results of the model experiments are to represent what actually happens on the full-scale ship. Without these precautions incorrect inferences can easily be drawn from the model results. The cross-section of the air-stream at the working section must be large enough to enable a model of reasonable size to be accommodated without serious disturbance of the flow, the velocity distribution across the working section must be uniform, the air-stream must be steady, and all large eddies and disturbances must be removed before the air reaches the working section. In order to achieve these and other necessary conditions much care has to be taken in the design of

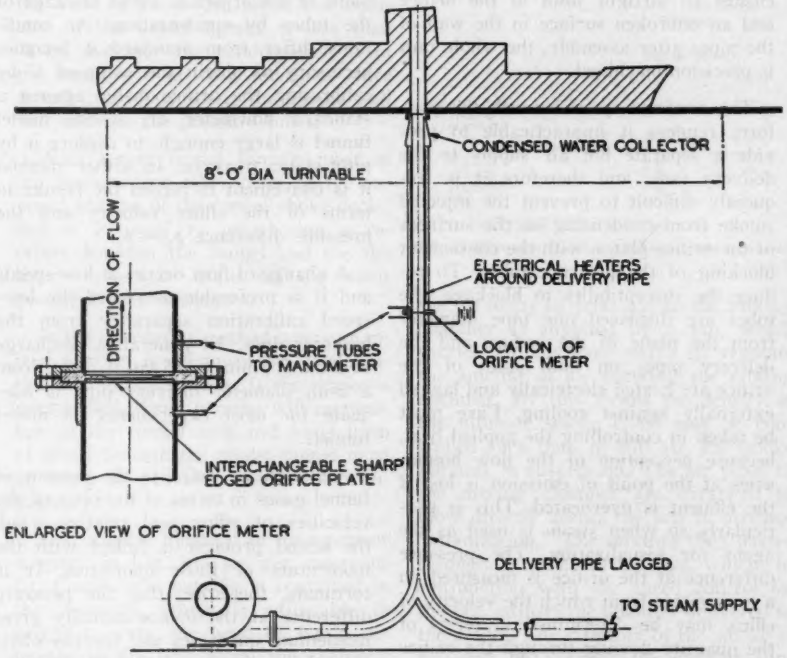


FIG. 17.—DIAGRAMMATIC SKETCH OF "SMOKE" SUPPLY TO MODEL FUNNEL

the wind tunnel; and any attempt to get results with crude, simple apparatus is strongly to be deprecated.

The Model. The dimensional limitations of the wind tunnels restrict the overall length of the model to 8 ft., and the scale of the model, therefore, must be selected to suit that dimension. The method of construction is not important, nor is it necessary to reproduce the components and fittings in minute detail, but all obstructions and excrescences likely to have an influence on the flow of air over the model must be included. In general it is sufficient to represent them in block form to their overall dimensions.

The model is positioned in the wind tunnel on an 8-ft. diameter turntable rebated into a platform supported in the central horizontal plane of the working portion. Below the turntable the delivery pipes to the model funnels are each taken to separate fans and all to a common smoke supply. A diagrammatic sketch of the arrangement is given in Fig. 17. The flanged couplings connecting the sections of the delivery system immediately beneath the model are designed with sleeves to fit pipes of different diameters between ¾-in. dia. and 1¼-in. dia. Each flange is recessed to accommodate interchangeable orifice plates of different diameters, and, to

ensure an airtight joint at the orifice and an unbroken surface in the wall of the pipes after assembly, the whole unit is precision machined.

The confined space beneath the platform renders it impracticable to provide a separate hot air supply to the delivery fans, and therefore it is frequently difficult to prevent the injected smoke from condensing on the surfaces of the orifice plates, with the consequent blocking of the pressure tubes. To reduce the susceptibility to blockage, the tubes are displaced one pipe diameter from the plane of the orifice, and the delivery pipes on both sides of the orifice are heated electrically and lagged externally against cooling. Care must be taken in controlling the applied heat, because perception of the flow boundaries at the point of emission is lost if the effluent is overheated. This is particularly so when steam is used as the agent for visualization. The pressure difference at the orifice is measured on a manometer from which the velocity of efflux may be determined in terms of the quantity flowing through the orifice as given by the usual relationship:

$$Q = C A_2 g \sqrt{2\rho\left(\frac{p_1 - p_2}{1 - m}\right)} \quad (1)$$

where

$$m = \frac{A_2}{A_1} = \frac{\text{area of the orifice}}{\text{area of the pipe}},$$

ρ = density of the fluid,

$p_1 - p_2$ = pressure difference at the orifice

The coefficient of discharge C is dependent upon (a) the velocity distribution across the pipe, (b) the ratio of the diameter of the orifice to that of the pipe, and (c) the disposition of the pressure points upstream and downstream of the orifice.[4, 5]

When an effluent containing moist gases is discharged, it is an advantage to displace the pressure tubes from the plane of the orifice to avoid blockage of the tubes by condensation. As conditions differ from standard it becomes necessary to obtain the value of C by calibrating the orifice meter against a standard flowmeter or, if the model funnel is large enough, to explore it by pitot-static traverse. In either method it is convenient to record the results in terms of the efflux velocity and the pressure difference $p_1 - p_2$.

A change of flow occurs at low speeds and it is preferable to record the low-speed calibration separately from the higher values. In general, a discharge of 6 cu. ft./min. to 25 cu. ft./min. from a 2-in. diameter delivery pipe is adequate for most experiments on model funnels.

Although we refer to the descent of funnel gases in terms of the ratio of the velocities of efflux and relative wind, the actual problem is linked with the momentum of those quantities. It is fortunate, therefore, that the pressure difference at the orifice actually gives momentum quantities and thereby obviates the necessity of determining the density of the fluid passing through the delivery pipe.

From the relationship given in (1) it is apparent that, since A_2 is constant in any one test, the quantity Q is proportional to $\sqrt{\rho(p_1 - p_2)}$

or $$Q^2 \propto \rho\,(p_1 - p_2);$$

i.e. $$p_1 - p_2 \propto \frac{Q^2}{\rho}$$

$$\propto \frac{Q}{\rho} \times \rho V A$$

$$\propto Q V A$$

$$\propto \text{momentum} \times A$$

To produce a visible discharge from the model funnel either steam or atomized paraffin is injected into the delivery system at a point below the orifice meter. The choice depends chiefly upon

the scale of the model, for if the diameter of the model uptake is less than ⅞-in. diameter the condensation from the steam will prove a nuisance, since it will drain back along the uptake wall and obstruct the pressure tubes.

With larger model funnels most of the condensation takes place in the exposed portion of the funnel above decks, and it is usual to incorporate a receiver between the funnel and the delivery system to drain off the water produced by condensation.

In the alternative method where atomized paraffin is employed the condensation is appreciably less. If the smoke is over-cooled, liquid paraffin will accumulate at the funnel exit and some form of drain beneath the model funnel must be fitted to provide against this contingency.

A schematic diagram of an atomizer is given in Fig. 18 (see also reference 6). Commercial paraffin is contained in the reservoir, which is adjustable in height to vary the head of liquid delivered to the jets. From the reservoir the paraffin passes through a filter to the heater, and on the remote side of the heater the paraffin, now in the form of smoke, enters the mixing chamber via an array of jets. Air to the mixing chamber is supplied by a fan, which carries an auxiliary valve to regulate the amount of air entering the mixing chamber. By this means the density of the ultimate smoke cloud may be controlled such that, in still air, the smoke remains in suspension. The air pipes to the mixing chamber are arranged to blow directly across the jets to produce "atomization." The mixture then passes through a cooler to a valve, which, in conjunction with the speed control on the fan, adjusts the quantity delivered to the model. The intensity of the smoke is adjusted by varying the current supplied to the heater, an excessive heat produces a very thin smoke, while insufficient heat results in excessive condensation on the funnel wall.

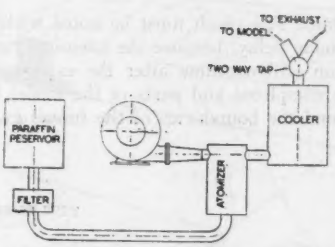


FIG. 18.—SCHEMATIC DIAGRAM OF PARAFFIN SMOKE GENERATOR

DETERMINING THE EXTENT OF DECK CONTAMINATION

The point of interest in the photographic records of a steam or smoke plume lies chiefly in the definition of the diffused lower boundary, in relation to its height over the deck corresponding to modifications of the ratio s/v. To study the concentration of the smoke at deck level it is necessary to employ a method which gives a direct indication of those parts of the deck actually contacted by the descending funnel gases. This is best achieved by a chemical reaction method. The technique which has proved satisfactory at the National Physical Laboratory requires the model to be sprayed with a white paint containing lead acetate* and a reagent introduced into the effluent.

Air only is discharged as the effluent, and after the velocity conditions have been adjusted to the required velocity ratio, sulphuretted hydrogen is injected slowly into the delivery pipe until the concentration over the deck is adequately defined by the resulting black

* For formula see Appendix

stain. The result must be noted without undue delay, because the chemical reaction will continue after the experiment is completed and parts of the model beyond the boundaries of the funnel gases may be affected also and produce a misleading impression of the extent of the concentration. The model must be cleaned and re-sprayed with the paint for each record.

THE EFFECT OF BUOYANCY

Another method of tracing the plume boundaries is to project the flow from the funnel on to a screen in the form of a shadowgraph and to sketch in the flow boundaries by hand. Because of the difficulties experienced in satisfying the similarity conditions with this technique, it has been found necessary to restrict these experiments to the detailed study of the buoyancy effects due to temperature as related to an unobstructed model of a funnel or chimney only. Increments in the height of the flow boundaries due to buoyancy are approximately in direct proportion to the difference between the free stream temperature and the funnel gas temperature, and also to the linear dimensions of the funnel uptake, but they are inversely proportional to the square of the relative wind velocity. Therefore, to represent the modification to the flow boundaries due to variations in temperature on a wind-tunnel model of the order of ⅛ in. to 1 ft. scale, the velocity conditions relative to normal cruising speeds would be of the order of 2 or 3 ft. per second. The difficulties of maintaining a uniform velocity distribution across the working section of the wind tunnel and within the model uptake are obvious, but, nevertheless, the mean values from a series of observations, for a particular set of conditions, give remarkably good agreement.

In this technique no visualizing agent is injected into the effluent. Air is taken from the reservoir of a compressor and fed through a low-capacity gas meter to the delivery system of the model (Fig. 19). The first section of the pipeline carries two 1-kilowatt heaters within the pipe, which heat the central core of the column of air passing along the pipe. The heated air then passes into a section of the pipe with a heating coil wrapped around it and extended as close to the point of emission as the model will allow. The heating stages thus separated provide a far better control of the temperature distribution across the pipe than would be possible with the heating elements located over the same section of the pipe. To minimize heat loss the whole of the delivery system is lagged to a point as close to the funnel exit as aerodynamic considerations will tolerate.

The temperature at any point of interest is measured in the normal manner by a copper-constantan thermocouple. At the funnel exit one thermocouple is attached to the wall of the uptake and another, with a sufficient length of lead to avoid wall-temperature interference, is positioned at the center of the uptake. The cold junctions of both thermocouples are positioned in the unobstructed free wind stream. The isothermals for a given velocity ratio and temperature difference, measured by a thermocouple exploration of the plume, are given in Fig. 20.

A number of experiments on model factory chimneys have been conducted with this type of apparatus, and from the results the mathematical deductions relative to the prediction of the flow boundaries from a factory chimney discharging into a turbulent air stream have been analyzed by Mr. L. W. Bryant, and reduced to coefficients in terms

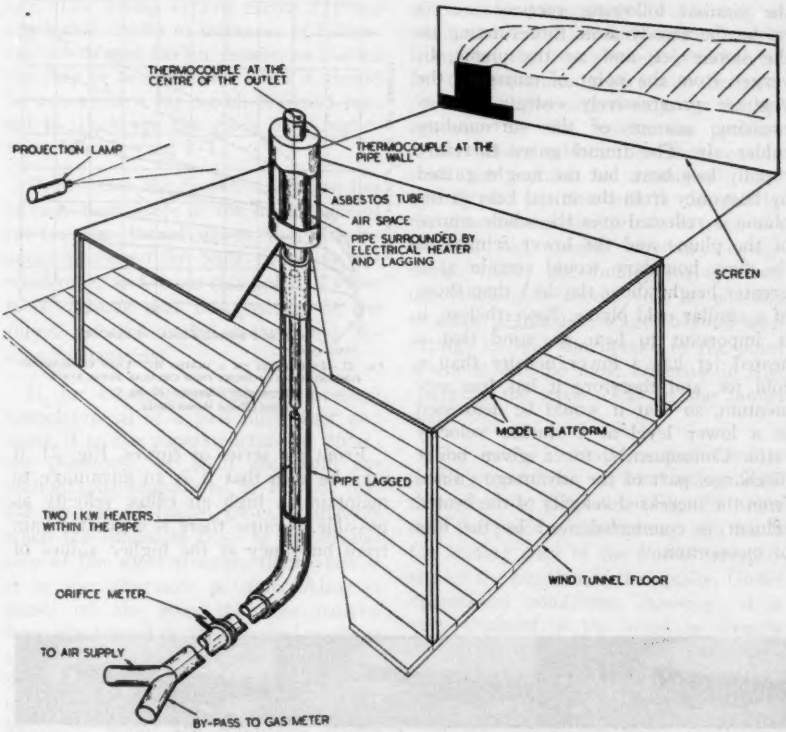


FIG. 19.—APPARATUS FOR EXPERIMENTS ON A HEATED JET

of the linear dimensions of the chimney and the velocity ratio s/v (see also reference 7).

As it is unlikely that ships' funnels will at any time be built high enough to be beyond the aerodynamic interference of obstructions on board the ship, the analysis from the heat experiments cannot be applied directly to the design of ships' funnels in general. Nevertheless, it is of value to the designer to be informed of the effects of temperature on the behavior of a column of smoke discharged into an air stream normal to its path. As the smoke impinges on the streamlines passing the funnel outlet,

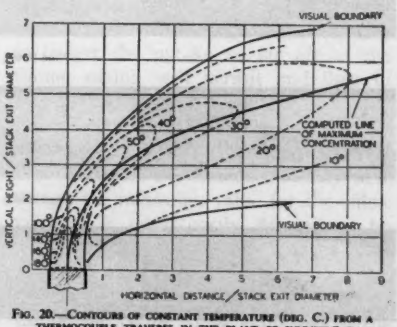

VISUAL BOUNDARY
COMPUTED LINE OF MAXIMUM CONCENTRATION
VISUAL BOUNDARY
VERTICAL HEIGHT / STACK EXIT DIAMETER
HORIZONTAL DISTANCE / STACK EXIT DIAMETER

FIG. 20.—CONTOURS OF CONSTANT TEMPERATURE (DEG. C.) FROM A THERMOCOUPLE TRAVERSE IN THE PLANE OF SYMMETRY OF A MODEL SMOKE JET
Efflux velocity/wind velocity = 2/1.
Temperature difference at exit = 200 deg. C.

the familiar billowing vortices are set up by the aerodynamic forces acting on the smoke jet, and, as the plume diverges from the point of emission, the vortices progressively entrain an increasing amount of the surrounding colder air. The funnel gases therefore rapidly lose heat, but the height gained by buoyancy from the initial heat in the plume is reflected over the whole course of the plume and the lower fringes of the flow boundary would remain at a greater height above the deck than those of a similar cold plume. Nevertheless, it is important to bear in mind that a heated jet has a lower density than a cold jet, and therefore it has less momentum, so that it would be depressed to a lower level at a similar velocity ratio. Consequently, for a given boiler discharge, part of the advantage gained from the increased velocity of the heated effluent is counterbalanced by the loss of momentum.

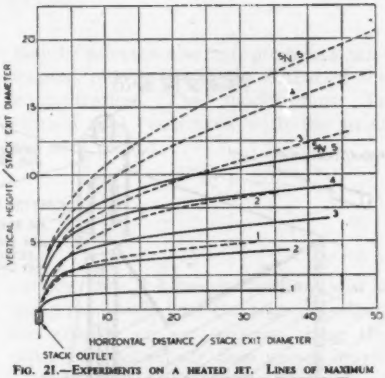


FIG. 21.—EXPERIMENTS ON A HEATED JET. LINES OF MAXIMUM CONCENTRATION OF SMOKE FROM CIRCULAR MODEL STACK
Temperature difference 180 deg. C.
Heated effluent shown dotted.

From the series of curves, Fig. 21, it may be seen that it is an advantage to maintain as high an efflux velocity as possible, because there is a greater gain from buoyancy at the higher values of

Wind ahead

Wind 10 deg. starboard

Wind 20 deg. starboard

Wind 30 deg. starboard

FIG. 22.—THE EFFECT OF FUNNEL POSITION
Efflux velocity/relative wind velocity = 2/1.

s/v. The values of s/v above 2/1 are applicable chiefly to instances of following winds and harbor conditions; when the ship is heading into wind it should be unnecessary for a well-designed funnel to discharge the gases at a higher velocity ratio than 2/1.

Positioning the Funnel. Mention has already been made of the importance of raising the funnel above the disturbances produced by bluff obstacles upwind of it, and, from this point of view, it is obvious that the position of the funnel relative to the obstacle is a design fundamental.

If, for example, we take an average funnel typical of a passenger liner and apply it to the superstructure in different positions, we shall see from the photographs reproduced in Fig. 22 that the descent of funnel gases is greater when the funnel is close to the frontal face of the superstructure than when it is in the aftermost position. Also, in winds off the bow, the flow pattern from the funnel in the forward position is appreciably more disturbed than when the funnel is partially screened by the breakaway from the superstructure. In following winds the conditions would be reversed, except those of velocity ratio which would improve with increasing ship speed.

However, both aesthetic considerations and practical requirements of the boiler-room arrangements are factors of importance, and either might outweigh the aerodynamic advantages of a differently positioned funnel.

Position of the Uptake within the Funnel. The flow boundaries for a number of funnel modifications have been sketched in the first part of this paper (Fig. 5). It must be accepted, however, that objections to a slavish adherence to aerodynamic efficiency must prevail when such a funnel would be out of keeping with the general lines of the ship, particularly as it is only at wind ahead and small deviations therefrom that the improvement is achieved. In circumstances where the designer prefers a slight rake to improve the appearance of the funnel the problem arises, where within the funnel casing shall the uptake be arranged? Frequently the position will be decided by the dimensional limitations of soot extractors and similar auxiliary equipment within the funnel casing; but, nevertheless, it should still be possible to allow a sufficient length beyond such fittings to lead the uptake to the point where it will be most effective in discharging the gases clear of the funnel wake.

The smoke boundaries and aerodynamic streamlines at wind ahead indicated on the diagrams (Fig. 5) show that the point of breakaway will be at the leading edge of the funnel irrespective of the position of the uptake. Under operational conditions, however, it is very doubtful if the wind is directly ahead for any appreciable part of a voyage, consequently this wind direction becomes of least importance.

Model experiments have shown that the worst conditions prevail when the ship is 20 deg. to 30 deg. out of wind, and, therefore, the shipbuilder should concentrate his efforts to obtain deck clearance over this range. The first step is to get the uptake as far aft as the funnel casing will permit and then to increase the efflux velocity until the lower fringes of the plume are dispersed clear of the disturbances in the immediate wake of the funnel. For any new design the minimum value for the efflux velocity is more easily determined by wind-tunnel experiments; in fact it is desirable that it should be so, because by this means it is possible to arrive at the power factor to be employed in the design of the forced draught and induced draught fans. By this approach

to the problem a sufficient reserve of power could be provided to give clear decks at all angles of wind.

It must be appreciated, however, that the wind-tunnel results are expressed in terms of the velocity ratio of efflux to relative wind, and it is for the designer to decide the figure for the speed of the natural wind to which he is prepared to power his fans.

METHODS OF PREVENTION

The Air Annulus. Model experiments on the effectiveness of an auxiliary air discharge around the funnel were initiated by the Admiralty in 1938, chiefly with a view to clearing the low-velocity region at the wall of the uptake and thereby preventing wisps of smoke from seeping down the funnel and entering the boiler-room ventilators. These experiments successfully dealt with that special problem, but they were not extended to deal with the main volume of smoke which probably descended to deck level at a more remote distance from the funnel.

The very small annular width which must be used in the model to correspond with full-scale dimensions requires the model funnel to be made to a high order of accuracy. The final portion of the uptake must be precision machined, and the annular spacing at the outlet must not only be concentric, but it must also be designed to give a good velocity distribution around its periphery. It is convenient to design the annular space leading to the outlet to the maximum width anticipated in the experiment, and thereafter to interchange separate nozzles to reduce the size of the annulus. The annular width ultimately decides the scale of the model, for it is not advisable to reduce that dimension below ⅛ in. If, however, this restriction results in the scale of the model being too large for the particular wind tunnel envisaged for the tests, then it is preferable to divide the experiments into two separate parts. First, a study would be made of the flow boundaries from a larger scale model of the funnel mounted on the superstructure only, and later the funnel arrangement giving the best results would be applied to a smaller scale model of the complete ship, when the whole plume would be recorded. On this model the actual annulus would be omitted and the resultant plume would represent the condition when the efflux velocity and annulus velocity were equal. If the boundaries of the plume are studied over a range of velocity ratios and superimposed on those obtained from the first part of the experiment, the descent of the funnel gases may be estimated for any wind condition within the limits bounded by the experiment.

It is not possible to arrive at an empirical formulae for the design of annular uptakes because the wide variations in ship design result in dissimilar streamlines in the plane of the funnel. The experiments which have so far been completed on this project indicate that the width of the annulus should not be less than 10 per cent of the uptake diameter. The velocity of discharge is dependent upon several factors, the chief of which is that of funnel height, for it is obvious that a high funnel would require a lower discharge velocity than a short funnel. Taking an average of the funnels of modern vessels, the annulus velocity should be between one and a half times and twice the velocity of the relative wind over the ship.

To provide a typical example, we will assume an uptake diameter of 5 ft., which, with a surrounding annulus 10 per cent of the uptake diameter, gives an overall dimension of 6 ft. diameter.

If the cruising speed is 15 knots and the natural wind 20 knots at wind ahead the relative wind speed over the ship will be 35 knots. At velocity ratios of 2/2 and 2/1 the quantity of air discharged from the annulus would be approximately 77,000 cu. ft. per min. and 102,000 cu. ft. per min. respectively.

The energy of the jet at the outlet would be 13.1 H.P. and 31 H.P. respectively, and the fans required to deliver that power would, in addition, have to overcome the drag losses in the trunking.

The full-scale application of the air annulus must be arranged in the initial stages of the funnel design, for it is generally difficult to find adequate space for a separate fan and trunking after the ship is built. Ideally, the fan should be controlled by an anemometer positioned as near as practicable within the undisturbed flow over the ship, so that the annular discharge is operated entirely according to the conditions of the relative wind. Such an anemometer would, of course, respond to small variations in wind speed and direction, but suitable damping could be provided in the fan speed-control system to restrict the fan so that it followed sustained conditions only.

Throats or Nozzles. To install an air annulus around the uptake of a ship already in service would probably entail such extensive modifications as to render it impracticable or too costly. A more convenient alternative would be that of fitting a throat or nozzle at the uptake outlet to reduce the area at the point of emission and thereby increase the efflux velocity in direct proportion to the areas of the uptake and the nozzle. It is important that the velocity distribution across the outlet should be as uniform as possible and this can only be achieved with an aerodynamically efficient nozzle. Two examples of a suitable nozzle are given in Fig. 23 "A," from which it may be seen that the sides of the nozzle require to be curved. This feature makes construction far more difficult than a straight-sided cone giving a similar contraction ratio, but the very much improved performance warrants the additional labor and expense. It will be appreciated, therefore, that a sharp contraction like that shown in "B," Fig. 23, should be avoided.

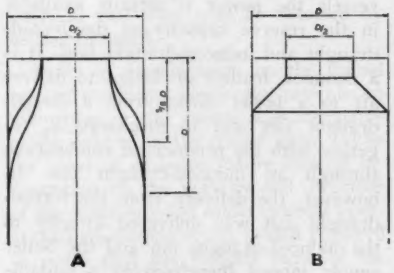


FIG. 23.—NOZZLE PROFILES

It must not be overlooked, however, that without the introduction of extra air into the uptake this device is most effective at full power only; at lower ship speeds the efflux velocity is relatively lower and the consequent plume from the funnel will be depressed closer to the decks.

A Proposed Solution to the Problem. As the results of wind-tunnel experiments have generally indicated the need of high efflux velocities to raise the funnel gases clear of the turbulent wake of the funnel and ship's superstructure, the designer is faced with the decision whether or not the nuisance value of the descending funnel gases is serious enough to warrant the installation of additional equipment to supply the power solely for that purpose. Hitherto the project has always been received very unfavorably, but the fact remains that mechanical power must be employed.

A study of contemporary boiler-room arrangements indicates that in many

vessels the power is already available in the reserve capacity of the forced-draught and induced-draught fans. It is a common feature of design to deliver air to a boiler direct from a forced-draught fan and to discharge it, together with the products of combustion, through an induced-draught fan. If, however, the delivery from the forced-draught fan was delivered directly to the induced-draught fan and the boiler supply tapped therefrom by a suitable valve (Fig. 24), a high efflux velocity

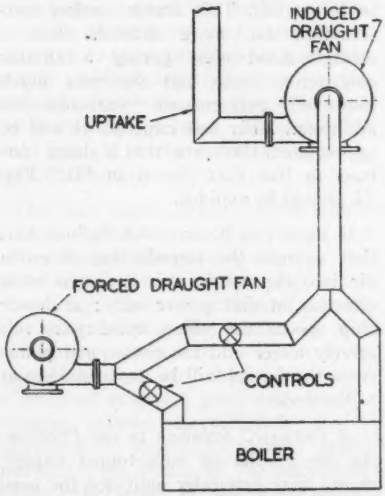


FIG. 24.—ARRANGEMENT TO GIVE A HIGH EFFLUX VELOCITY IRRESPECTIVE OF SHIP SPEED

could be provided at any time the relative wind conditions demanded it. This could be done independently of the ship speed, because the valve would be operated to divert only the required amount of air to the boiler. The balance between the speed of the two fans must be computed in terms of the relative wind speed and the ship speed, and a suitable chart prepared for the boiler-room engineers. This would entail the addition of an anemometer to the ship's equipment, and it would require to be positioned as nearly as practicable in the free stream over the ship. The anemometer would give a remote reading of the wind speed and direction on an indicator in the boiler room, and it would be the responsibility of the engineers to adjust the speed of the fans in accordance with those readings.

In those vessels where the air supply to the boiler is obtained by increasing the boiler-room pressure above atmospheric, the solution is equally possible. A controlled inlet from the boiler room leading directly into the uptake will also produce an efflux velocity to give clear deck conditions within the limits of the design.

It follows, therefore, that either of these methods will afford the designer a means of obtaining the equivalent of full-power efflux conditions at all times and at any ship speed. When the ship is in harbor and both the relative wind speed and the ship speed are zero, nothing will keep the decks free from soot deposits other than a large asbestos umbrella.

Model and Full-Scale Comparison. Manifold difficulties are encountered when attempting to record the pattern of the smoke plume under full-scale conditions. An escort of equal speed to that of the parent ship is required, from which the photographs can be taken, and this vessel must also carry a stabilized or articulated framework to support the camera. On board the parent ship other observers are needed to record the efflux velocity of the funnel gases, the speed and course of the ship, and the velocity and direction of the relative wind. Cooperation by signal between the two ships must be organized in order to ensure that they will be on parallel courses, at similar speeds, when the records are being taken. The fact that the whole operation requires to

be conducted as a separate experiment introduces further complications in terms of time and cost. It is seldom possible to undertake these experiments during the acceptance trials of a vessel because of the many other tests on the ship's equipment then in operation, nor is it convenient to attempt such records when the ship is in commission because of the inconvenience to passengers.

Difficulty also arises in positioning the camera to photograph the plume from a station on board the ship herself. When the camera is trained aft it photographs the smoke astern as well as the smoke over the ship, so that it becomes quite impossible to determine where, on the ship, the smoke reaches the deck level. When the camera is positioned aft and trained forward, the identification of specific parts of the plume is equally impossible. It becomes very much more convenient, therefore, to resort to visual observation, and many practical aids to this method may be devised. There is also the fact that the lower fringes may be scarcely visible, but they are usually above atmospheric temperature, while the sulphur dioxide content gives them a distinct odor, so that they may be detected by feeling or smell.

Areas of discoloration on the funnel and surrounding deckworks provide an indication of the extent and frequency of the descent of the funnel gases below the point of emission. More concise information on the velocity and direction of the relative wind under normal operational conditions at sea is still required in order that the velocity ratio of the model experiments may be adjusted to them.

We at the National Physical Laboratory are very much indebted to some of the leading shipping companies, whose kindly cooperation over the past few years has afforded us the opportunity of gaining a wider knowledge of full-scale smoke conditions. Much useful information has been obtained; and the impressions gained from this experience lead to the conclusion that the flow boundaries indicated by model experiments approximate very closely to those full scale.

ACKNOWLEDGMENT

The above work has been carried out in the Aerodynamics Division of the National Physical Laboratory and this paper is published by permission of the Director of the Laboratory.

The author wishes to acknowledge his indebtedness to Mr. C. F. Cowdrey, for his valuable assistance with the experimental work and in the application to full-scale operation.

References

(1) L. W. Bryant and D. H. Williams: "Discontinuous Flow Around the Edge of a Bluff Obstacle," Aeronautical Research Committee, R. and M. No. 962 (January 1925).

(2) R. W. Nolan: "Design of Stacks to Minimize Smoke Nuisance," "Trans. Am. Soc. N.A.M.E.," 54 (1946), p. 42.

(3) J. Valensi and L. Guillonde: "Sur les Formes de Carénage de Cheminées de Navires Propres à Eviter le Rabattement des Fumées," "Bulletin de l'Association Technique Maritime et Aéronautique, 47 (1948), p. 173.

(4) E. Ower: "The Measurement of Air Flow" (Chapman and Hall).

(5) British Standards Institution: "Flow Measurement," British Standard Code B.S. 1042: 1943.

(6) C. Salter: "A Multiple Jet White Smoke Generator." (To be published in the R. and M. series.)

(7) L. W. Bryant: "The Effects of Velocity and Temperature of Discharge on the Shape of Smoke Plumes from a Funnel or Chimney: Experiments in a Wind Tunnel" (Adm. 66 unpublished).

Appendix

China Clay Solution for Chemical Reaction Experiments

China clay	400gms.
Lead acetate	80 gms.
Acetone	1,200 c.c.
Glycerine	20 c.c.
Water	1,000 c.c.

Dissolve the lead acetate in the water.
Add to china clay.
Add the acetone and glycerine.
The above quantities produce 1 winchester of the solution.

BOOK REVIEW

"FUNDAMENTALS OF ACOUSTICS"

By Lawrence E. Kinsler and Austin R. Frey

Published by John Wiley and Sons, Inc., 516 pages—$6.00

Reviewed by Murray Strasberg
Physicist, Bureau of Ships

No longer can the Naval engineer consider acoustics as an esoteric subject to be ignored. Developments during and since the Second World War have made acoustic requirements increasingly important in the design of Naval vessels.

The Naval engineer's interest in acoustics results primarily from the problems of noise and vibration reduction. In preliminary design, the ship designer has been forced to consider such unorthodox questions as whether a single- or twin-screw vessel is inherently quieter, or whether modifications in hull form will result in a reduction of underwater noise. In the detailed design, he has been required to design a propulsion system without reduction gears, to determine whether four- or five-bladed propellers have less vibration, or to choose between Waterbury and IMO pumps on the basis of noise.

Excessive vibration causes discomfort and results in fatigue failure of components. Airborne noise interferes with communications, and impairs the efficiency of personnel. Noise transmitted to the water can be used by the enemy for detection and for attack with acoustic undersea weapons. Waterborne and structure-borne noise interferes with the operation of the ship's own acoustic detection equipment. For all these problems, there are two types of solution; either to reduce the noise or vibration at its source, or to reduce the transmission.

There are a number of routine and "rough and ready" procedures for accomplishing noise reduction. Unfortunately, however, noise sources and transmission paths often behave contrary to the dictates of common sense, so that a basic knowledge of acoustic principles is desirable for the correct application of noise-reduction procedures.

A study of "Fundamentals of Acoustics," by Lawrence Kinsler and Austin Frey, will provide the required basic knowledge. The first four chapters follow the traditional sequence, probably first introduced by Lord Rayleigh, of discussing simple-harmonic motion, vibrating stretched strings, longitudinal vibrations of bars, and transverse vibrations of bars, stretched membranes, and plates. The next five chapters complete the discussion of classical theoretical acoustics, covering

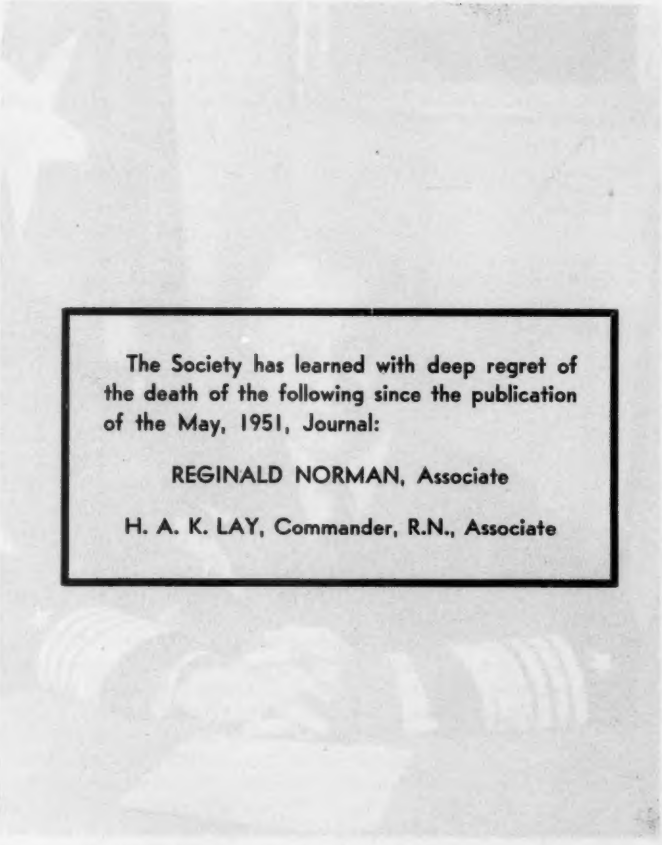
the transmission of sound in one direction, the spherical propagation of sound, acoustic resonators and filters, and the absorption of sound. The last six chapters discuss various problems in applied acoustics: loudspeakers, microphones, physiological and psychological acoustics, room acoustics, underwater sound, and ultrasonics.

Although there are many textbooks on the subject, the Kinsler and Frey work may well fill a gap in the existing literature between the primarily descriptive books and the more advanced mathematical texts. All the subject matter is presented from a theoretical viewpoint, using analytic methods and concepts. However, the discussion is couched in simple language, and only enough mathematics is used to provide a firm foundation for understanding. Accordingly, the not-too-mathematical reader will undoubtedly prefer this book to the more advanced texts.

The above is not to imply that little mathematical background is required. The discussion in the theoretical chapters presumes a knowledge of the use of complex-variable notation, of Fourier series, and some familiarity with Bessel functions. From a pedagogic viewpoint, it might have been desirable to provide some discussion of these subjects. Too many engineers will require preliminary instruction, using some other text, to provide them with the required background.

The increased number of scientists working in acoustics during the past years has resulted in the development of many new concepts in an essentially-classical field. The book is written in terms of these new concepts, and many of the newer terms which do not appear in the older texts are defined here: for example, recognition differential, articulation index, transmission anomaly, and directivity index.

The engineer looking for the quick solution to his noise problem will not find his answer here. To such a casual reader, it is hoped that the book will indicate that acoustics is a reasonably complicated subject and that he is not an acoustics expert simply because he has a good pair of ears. For the serious reader, interested in a firm foundation in the subject, "Foundations of Acoustics" is highly recommended.



The Society has learned with deep regret of the death of the following since the publication of the May, 1951, Journal:

REGINALD NORMAN, Associate

H. A. K. LAY, Commander, R.N., Associate

ADMIRAL FORREST PERCIVAL SHERMAN, U. S. NAVY

ADMIRAL FORREST PERCIVAL SHERMAN, U. S. NAVY

It is with deep regret that the Society reports the sudden death of ADMIRAL FORREST PERCIVAL SHERMAN, U. S. Navy, Chief of Naval Operations, at Naples, Italy, on 22 July 1951, while on a foreign mission in connection with his duties as a member of the Joint Chiefs of Staff.

Forrest Percival Sherman was born in Merrimack, New Hampshire, on October 30, 1896, son of Frank James and Grace Allen Sherman. He was graduated from Melrose High School, Melrose, Massachusetts, and had entered the Massachusetts Institute of Technology when he was appointed to the U. S. Naval Academy in 1914. While a Midshipman he won the Navy "N" as a member of the intercollegiate championship fencing team, was regimental adjutant, and was awarded the Class of 1871 Prize, Navy dress sword and knot, as the member of the graduating class most proficient in practical and theoretical ordnance and gunnery. Graduated with distinction, second in the Class of 1918 (199 members) and commissioned Ensign on June 28, 1917, he subsequently attained the rank of Rear Admiral on April 3, 1943, and Vice Admiral on December 28, 1945. He took the Oath of Office as Chief of Naval Operations, with the accompanying rank of Admiral, on November 2, 1949.

During World War I, he served in Mediterranean waters in the USS *Nashville*. He later served in the destroyer *Murray*, which was based on Brest, France, and in August 1919 joined the USS *Utah*. A year later he was transferred to the USS *Reid*, flagship of Commander Destroyer Squadron One, Atlantic Fleet. He commanded the USS *Barry* from May to October 1921, when he reported for duty as Flag Lieutenant to Commander Control Force, Atlantic Fleet. He served for eight months in that assignment attached to the USS *Florida*, flagship.

In June 1922 he reported to the Naval Air Station, Pensacola, Florida, for flight training. Designated Naval Aviator on December 22, 1922, he joined Fighting Squadron TWO, Aircraft Squadrons, Battle Fleet, in April 1923. A year later he returned to Pensacola to serve as an instructor until June 1926, and the following year had instruction at the Naval War College. Reporting in June 1927 to the USS *Lexington*, then fitting out, he served in that aircraft carrier from her commissioning, December 14, 1927, until December 1928. He then joined Scouting Squadron TWO, based on the aircraft carrier *Saratoga*, and was placed in command of that squadron in April 1929. In May he was assigned duty as Flag Secretary on the staff of Commander Aircraft Squadrons, Battle Fleet, USS *Saratoga*, flagship, and remained in that assignment until June 1930.

He was an instructor in the Department of Seamanship and Flight Tactics at the Naval Academy during the school year 1930-1931. In May 1931 he rejoined the *Saratoga* with duty on the staff of Commander Aircraft, Battle Force, U. S. Fleet. In June 1932 he assumed command of Fighting Squadron ONE, based on the *Saratoga*. That squadron won the Aircraft Gunnery Trophy in 1932-1933, and he was commended by the Secretary of the Navy for the high efficiency of his squadron. He personally won the Navy "E" in both dive bombing and fixed guns.

In June 1933 he reported for duty in the Bureau of Ordnance, Navy Department, Washington, D. C., where he had charge of the Aviation Ordnance Section until June 1936. After a year's service as Navigator of the USS *Ranger,* he was transferred to duty as Fleet Aviation Officer on the staff of Commander Battle Force. He continued staff duty from January 1938 until February 1940 on the staff of Commander in Chief, U. S. Fleet.

During his tour of duty, ending February 1942, in the War Plans Division. Office of the Chief of Naval Operations, Navy Department, he was a member of the Permanent Joint Board on Defense, Canada-United States, and also served as Naval Aviation Advisor at the Atlantic Conference in August 1941. Following three months' duty at Headquarters, Commander in Chief, U. S. Fleet, as a member of the Joint Strategic Committee, he assumed command of the USS *Wasp* in May 1942, and was in command of that carrier when she was sunk by enemy action in the Solomon Islands area on September 15, 1942.

For "extraordinary heroism as Commanding Officer of the USS *Wasp* and Flag Captain to the Commander of a Task Force during the occupation of Tulagi-Guadalcanal and subsequent operations . . . ," he was awarded the Navy Cross. The citation continues: "In addition to supervising plans for employment of aerial support to cover the initial landing, Captain Sherman, by his proficient skill and astute leadership, contributed to the relentless fighting spirit and aggressive courage which enabled his group to destroy hostile opposition in the air and on the beach. His outstanding performance of duty was a decisive factor in the success of our forces and was in keeping with the highest traditions of the United States Naval Service."

Following the sinking of the *Wasp,* he was designated Chief of Staff to Commander Air Force, Pacific Fleet. He was awarded the Legion of Merit for "exceptionally meritorious conduct (in that capacity) from October 14, 1942, to November 24, 1943. . . ." The citation states further: "A skilled and aggressive leader, (he) coordinated the planning, training and logistic requirements which produced efficient naval aviation in the Pacific throughout a vitally important period. His resourceful initiative, keen foresight and conscientious devotion to duty were contributing factors in the advanced state of preparedness of the aviation units of the Pacific Fleet for the Central Pacific Campaign and the successful prosecution of the war against the enemy."

On November 10, 1943, he was transferred to duty as Deputy Chief of Staff to Commander in Chief, Pacific Fleet and Pacific Ocean Areas. He was awarded the Distinguished Service Medal for "exceptionally meritorious service . . . (in that assignment) from November 10, 1943, to September 2, 1945. . . . " The citation points out: "Rear Admiral Sherman rendered distinguished service in the preparation of plans and the coordination of our campaigns to bring about the early capitulation of the Gilberts, Marshalls, Marianas, Western Carolines, Iwo Jima and Okinawa, our final attacks on Japan, and the naval phases of the occupation of the defeated Empire. As head of the War Plans Division on the staff, he contributed essentially to the successful prosecution of the war in the Pacific and his unfailing efforts were of vital importance in forcing the enemy to surrender. . . ."

Representing the Navy in the initial conferences with the Japanese at Manila in August 1945, he was present aboard the USS *Missouri* in Tokyo Bay when the formal surrender of the Japanese was signed on September 1, 1945 (EST). In

October 1945 he was ordered to duty as Commander Carrier Division ONE, and in December of the same year became Deputy Chief of Naval Operations (Operations), Navy Department, Washington, D. C., where he was responsible for plans, fleet operations and training, intelligence and joint activities.

In January 1948 he was designated Commander U. S. Naval Forces, Mediterranean, which title was changed on June 1, 1948, to Commander Sixth Task Fleet, and served in that assignment until appointed Chief of Naval Operations. He assumed the duties of that office, in the rank of Admiral, on November 2, 1949.

In addition to the Navy Cross, the Distinguished Service Medal, the Legion of Merit, and the Purple Heart Medal (awarded for wounds received aboard the USS *Wasp*), Admiral Sherman has the Victory Medal, Patrol Clasp; the American Defense Service Medal, Fleet Clasp; the Asiatic-Pacific Campaign Medal; the American Campaign Medal; the World War II Victory Medal; and the Navy Occupation Service Medal.

The Government of Great Britain made him an Honorary Companion of the Order of the Bath and awarded him the Oak Leaf Emblem after Mention in Despatches. The King of Greece awarded him the Grand Cross of the Order of the Phoenix; the Government of Italy the Grand Cross of the Military Order of Italy; and the Government of France the degree of Grand Officer of the Legion of Honor.

Admiral Sherman was an Honorary Member of the Society.

The Society has to announce with much pleasure that the following have been admitted to membership since the publication of the May, 1951, issue.

NAVAL

Andrew, Mario Iacobucci, Lieut. Commander, U.S.N.R.,
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Grove, Michael Motte, Ensign, U.S.N.,
USS *Renshaw* (DDE 499) % Fleet P. O., San Francisco, Calif.

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TRANSFERRED ASSOCIATE TO CIVIL

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RESIGNED

Anderson, W. L., Captain, U.S.N., Naval

ASSOCIATION NOTES

ORIGINAL ARTICLES

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $40.00 to $150.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be mailed addressed to:

Secretary-Treasurer
American Society of Naval Engineers
605 F St., N. W.
Washington 4, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which insufficient material is already on hand.

SUBJECT MATTER AND AUTHORS

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries.

(a) What subject would you, as a member, like to see covered by an article in the JOURNAL.

and

(b) Who do you consider best qualified to prepare such an article authoritatively, and what is his address (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

PHOTOGRAPHS

Photographs of current or historical interest to readers of the JOURNAL are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

NOTICE OF DEATHS

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

SOCIETY LAPEL BUTTONS

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold background.

It is available to all members at fifty cents (50c) each.

LIFE MEMBERSHIPS

Any member may now purchase a life membership under the following rules:

(a) Any dues paid for the year in which life membership is purchased will be credited to the cost of the life membership.

(b) Life memberships are non-transferable and terminate with the death of the member.

(c) No refund will be made on account of death or resignation of a life member.

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Under 25 years	250.00

(e) All life memberships paid in shall be protected by a transfer on the books of the Society of Series G, U. S. Government 2½ per cent bonds. Thus a life membership will be established. It will be credited with moneys paid in for life memberships and debited with the regular amount of annual dues for each life member each year.

LOCAL CHAPTERS

Local Chapters of the American Society of Naval Engineers may now be authorized by the Council. The following rules for recognition of Local Chapters have been adopted by the Council:

(a) Twenty or more regular members, Naval or Civilian, may apply for a Charter as a Local Chapter.

(b) Each application must be accompanied by a list of members and a copy of local By-laws. The latter must meet the following minimum requirements.

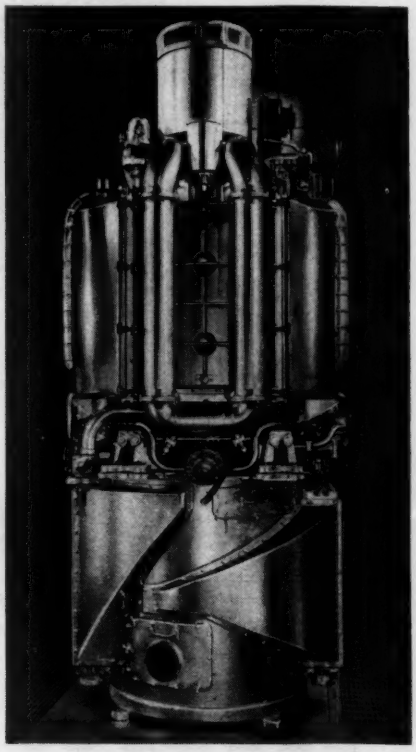
1. All financial dealings of the local Chapter must be independent of the Society and can in no way obligate the Society as a whole. The local chapter shall set up on a non-profit basis. All accounting shall be local.

2. Each active local Chapter shall receive from the Society a grant of $0.50 per year for each naval member, civil member or associate who resides in the local area and who associates himself with the local Chapter. This grant may be used by the local Chapter to defray any necessary expenses, including (a) Letterheads, (b) Notice cards, (c) Postage, (d) Minimum rental allowance, (e) Secretarial service (part time).

3. Any technical papers presented at local chapter meetings, shall be submitted to the Secretary-Treasurer of the Society for consideration for publication in the JOURNAL. Any paper accepted will be paid for at regular rates.

4. Each local chapter shall submit a quarterly report of its activities to the Secretary-Treasurer. These reports must reach Washington, D. C. prior to 15 January, 15 April, 15 July and 15 October. This report shall contain a list of active local members as of the first day of the quarter for the purpose of computing the amount of grant.

5. Associate members of the Society shall be eligible for membership in a Local Chapter, entitled to all the privileges of other members except voting and holding office.



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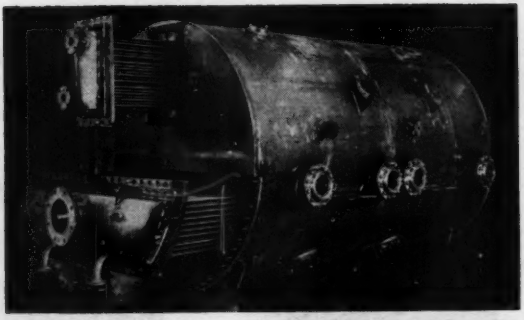

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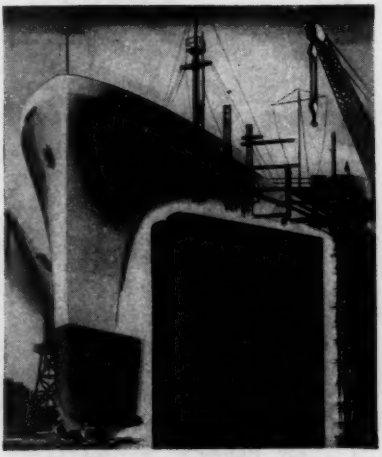

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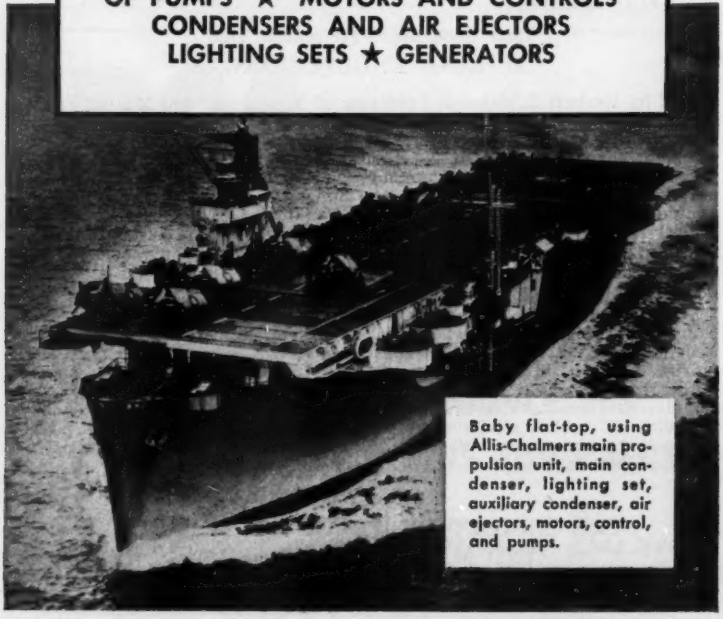

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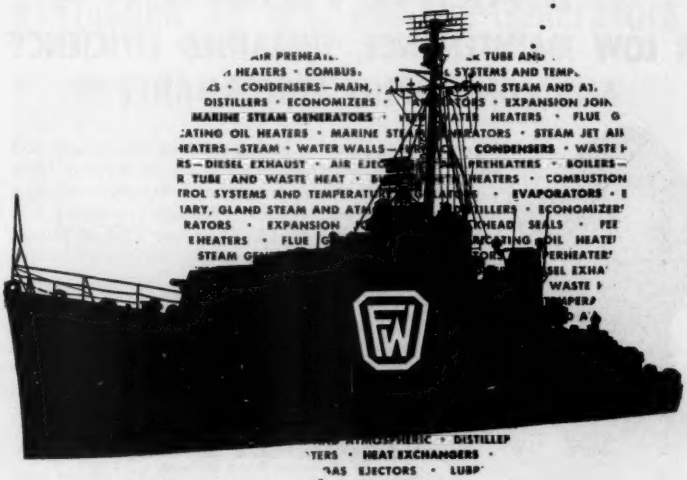

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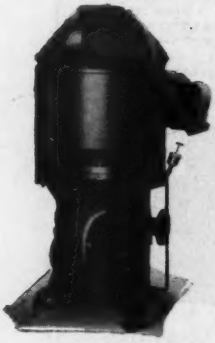

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200-83F

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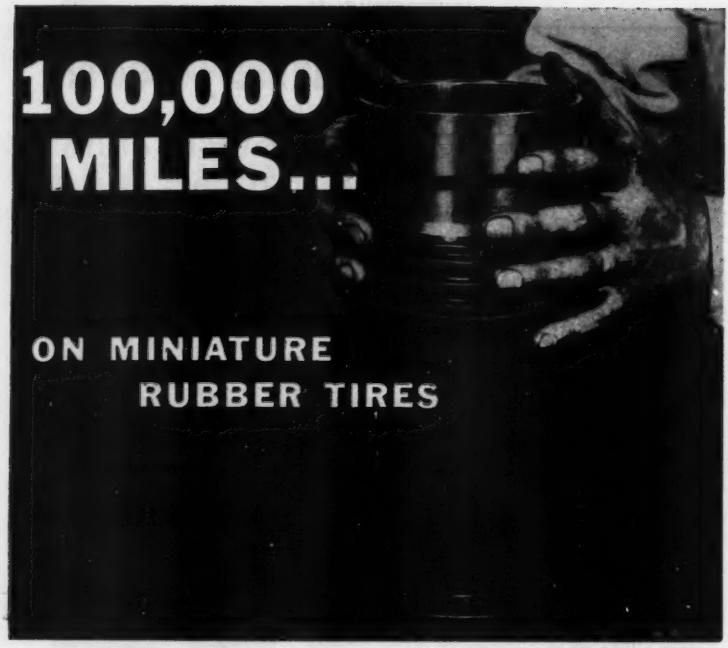

Photograph courtesy of Cummins Engine Company, Inc., Columbus, Indiana

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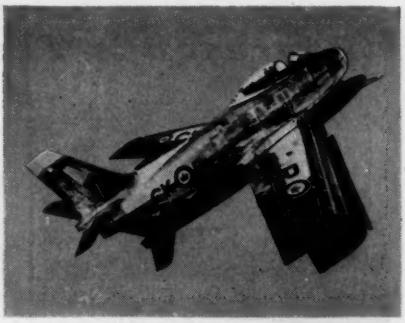

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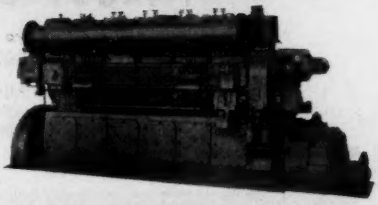

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